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REVIEW OF ENGINEERING PROGRESS ABROAD

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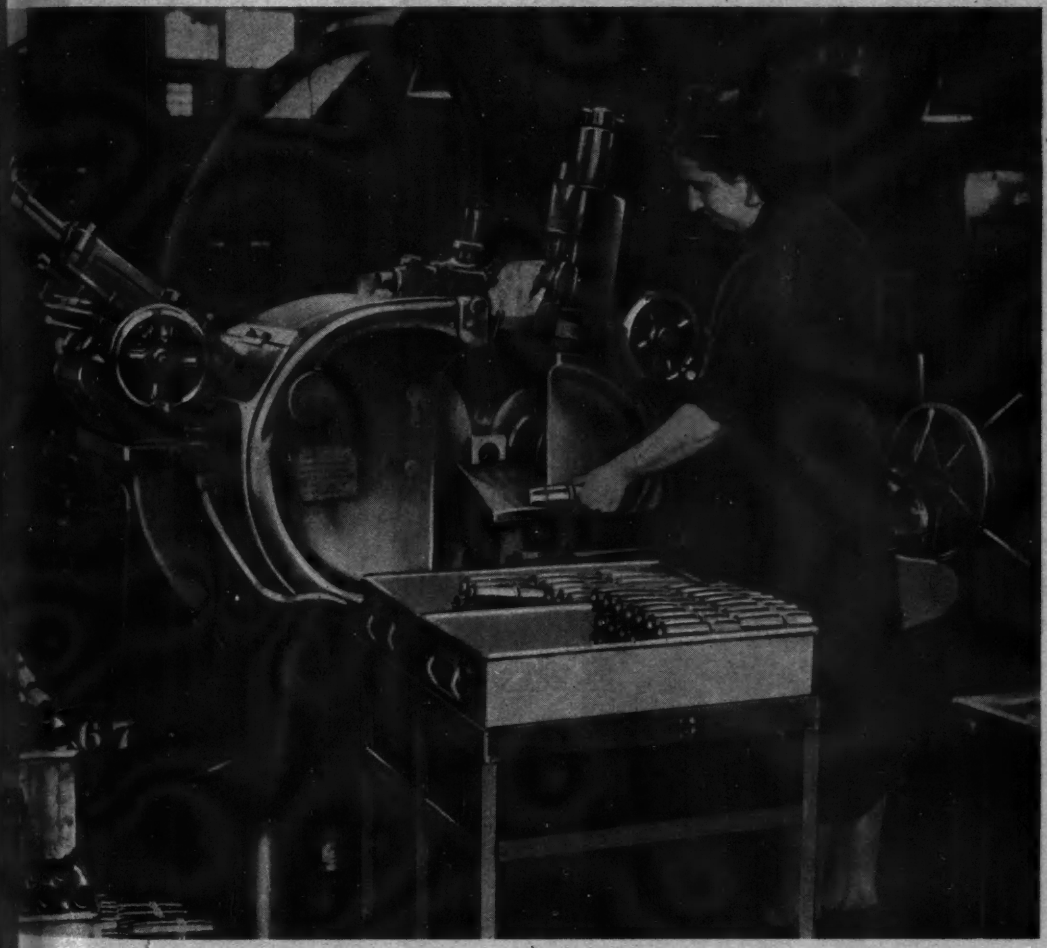

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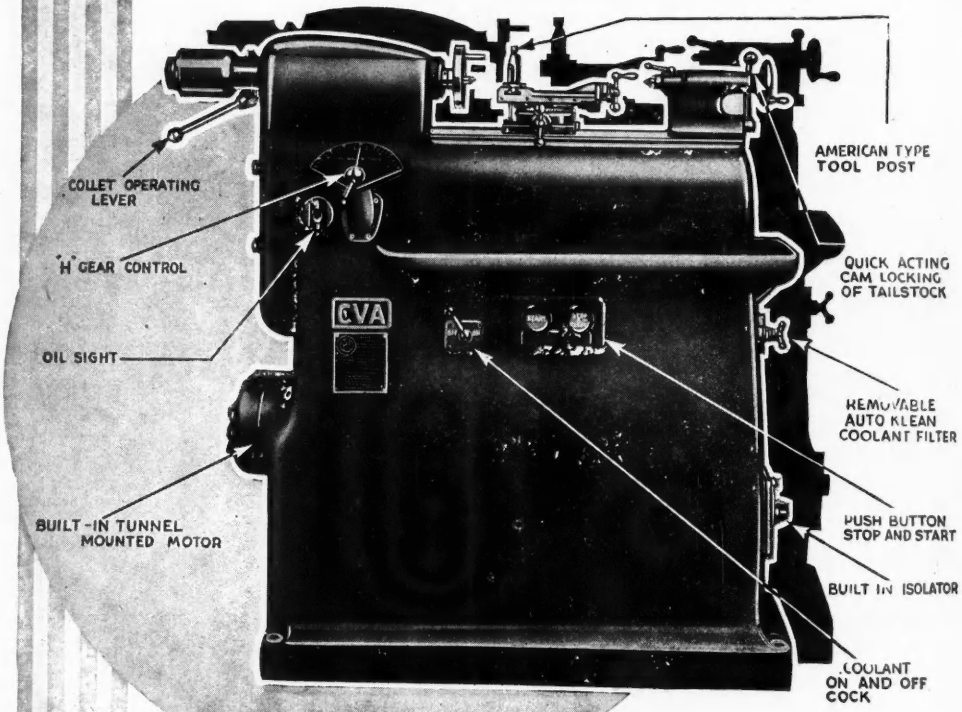

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CALCULATION OF STIFFENED SHELLS IN METAL AEROPLANE DESIGN.

By DR.-ING. E. SCHAPITZ VDI, Seestadt Rostock. (From *Zeitschrift des VDI*, Vol. 86, No. 33/34, 22nd August, 1942, pp. 497-507.)

The Features of Shell Structures.

A SHELL is a thin-walled structure—stiffened as far as necessary—which allows, as far as possible, an even distribution of direct and shear forces. This type of structure, as exemplified on an aerofoil section, is shown in Fig. 1. In contradistinction to it, Fig. 2 shows a wing section which is representative of a system in which the shear is carried by shear webs and the direct forces are concentrated in the flanges (longitudinals).

In general, both these types are being built in duralumin, with the possible exception of floats and boats. The reinforcing sections (stringers and stiffeners) are made of sheet material, or extruded, or rolled.

The Special Strength Requirements in Aeroplane Design.

In the air, the aeroplane structure is subjected to aerodynamic forces; when landing, it is subject to the landing impact. For each loading case the "safe load" is that which is very seldom likely to be exceeded during the life of the aeroplane. The German design specifications require for the "safe load" a safety factor of 1.35 against permanent deformation, and 1.8 against rupture. There are, in addition, some requirements as to stiffness, e.g., with regard to vibration. The change-over in design from the framework type to the shell type or monocoque, necessitated a change in the stress analysis. After a transition period in which stressing has been conducted in a rather tentative way, research succeeded in creating the necessary fundamentals. For checking the calculations, loading tests on complete structures are still indispensable.

The Problems of Stress Analysis in Metal Aircraft Design.

For weight reasons it is not generally possible to make the skin so thick as to remain unbuckled up to the load causing failure of the whole structure. As, however, buckles caused by low multiples of the buckling load disappear when unloading, this is not even necessary. The skin must be made just so thick that after releasing the 1.35 fold "safe load" no buckles remain. However, to estimate the load-carrying capacity one must be able to calculate, at least approximately, the redistribution of stresses and secondary stresses after buckling occurs.

A further difficulty is encountered by load transfer from locally-applied loads (diffusion). The elementary theory of bending and torsion assumes a continuous decrease of forces at one end of the shell; this is here confronted with a massing of single forces at the other end. With thin-walled shell structures the principle of St. Venant is not applicable as the aforesaid process is not localised but rather extensive, so that at points of application of concentrated forces great bulk is required which thus considerably influences the weight of the structural part. Figs. 3 to 5 show that load transfer, as affected by cut-outs, is also a diffusion problem.

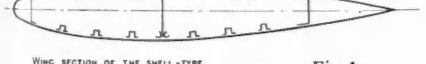


Fig. 1

Fig. 2

The stress analysis of metal aircraft can be subdivided into the following groups:—

I. Stability conditions.
II. Stiffened plates and shells subsequent to buckling of the skin.
III. Elementary theory of stiffened cylindrical and conical shells.
IV. Diffusion problems.

Stability Conditions.

In determining the buckling stress it is assumed that the flexural stiffness in all directions is the same (isotropic behaviour), whilst in investigating the strength of whole panels the stiffness of reinforcements perpendicular to each other must be taken into account (orthotropic behaviour).

The main types of loading of a panel are shown in Fig. 6; the edges can be free, supported or fixed (clamped). In case of a supported edge, the edge is restrained against deflection, but free in inclination. With a fixed edge, both deflection and inclination of the edge are restrained. In all problems of plate stability it is of the greatest importance to clarify the influence which initial buckles (initial curvature) have on the buckling stress. Differences, due to initial buckles, between theoretical buckling stress and test results may be appreciable. Table I gives a short summary of the most common buckling formulas used in metal aircraft stressing. The indices P, O, R refer to flat plate, curved plate and cylindrical shell respectively. For combined compression and shear the values σ_K, τ_K refer to the critical stresses. Either one of the two stresses or their ratio must be known. The formula given in Table I has been derived by the energy method from H. Wagner. The length l refers to a circular cylindrical shell and has influence on the compression buckling stress only if $l \ll r$, whereby the shell can be assumed reduced to a laterally compressed plate, or when $l \gg r$, whereby the shell approaches the form of a column. The coefficient 0.3 takes initial buckles into account (the theoretical value is 0.602).

Stiffened Plates and Shells subsequent to Buckling of the Skin.

Two essential cases of buckling, those under pure compression and pure shear, will be dealt with in this chapter. Buckling of plates due to combined compression and shear will not be considered.

Take, for example, a structure in which flat or curved plates have been compressed just beyond their buckling limit (Fig. 8). The shortening of the plate in direction of the load is constant over the width of the plate. It can be divided into two parts: the elastic (proportional to the compression stress) and the geometric, which is due to the difference between arc and chord, and becomes greater the deeper the buckles.

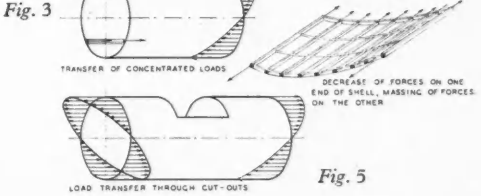

DIFFUSION PROBLEMS
TRANSFER OF CONCENTRATED LOADS
DECREASE OF FORCES ON ONE END OF SHELL, MASSING OF FORCES ON THE OTHER
LOAD TRANSFER THROUGH CUT-OUTS

Fig. 3 *Fig.* 4 *Fig.* 5

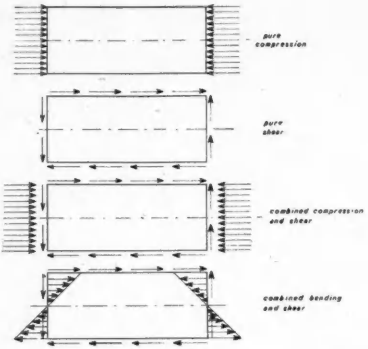


Fig. 6. Plates under various types of loading.

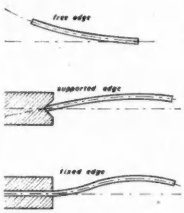


Fig. 7.
Edge restraint of plates.

As the buckles are deepest in the middle of the plate the elastic shortening, and thus the stress there, is the least. This fact can be taken into account by the conception of the effective width. Instead of the full width of the plate b, a part b_m of it is thought as carrying the stress σ_L of the stiffeners (Fig. 8). If we denote by σ_x the average plate stress, and the load carried by the plate (skin only) P_H, the following relation can be written down $P_H = \sigma_x b_3 = \sigma_L b_m s$, and thus

$$\frac{b_m}{b} = \frac{\sigma_x}{\sigma_L} \qquad .. \quad .. \quad (1)$$

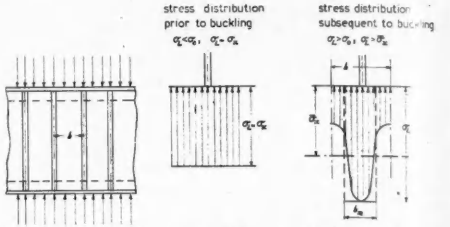


Fig. 8. The effective width of buckled plates in compression.

For given plate dimensions the ratio b_m/b is dependent on the ratio σ_L/σ_O, the ratio of stiffener stress to buckling stress.

For the flat plate Marguerre has calculated the approximate formula

$$\frac{b_m}{b} = \sqrt[3]{\frac{\sigma_O}{\sigma_L}} \qquad .. \quad .. \quad (2)$$

A diagram for the determination of the effective width of both flat and curved plates is shown in Fig. 9.

The parameter involved is $\lambda = \frac{\sigma_R}{\sigma_P}$, whereby σ_R is the compression buckling stress of a cylindrical shell of the same radius and σ_P the buckling stress in com-

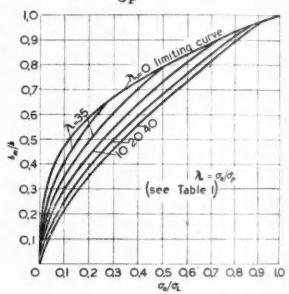


Fig. 9
Diagram for the determination of the effective width of flat and curved plates.

TABLE I. BUCKLING FORMULAE FOR ISOTROPIC PLATES AND SHELLS.

LOAD	FLAT PLATES	CURVED PLATES	UNSTIFFENED CIRCULAR CYLINDRICAL SHELLS
COMPRESSION	$\sigma_P = k'_P \frac{E}{1-\nu^2}\left(\frac{s}{b}\right)^2$ for fixed edges $k'_P \approx 6{\cdot}0$ for supported edges $k'_P \approx 3{\cdot}3$ if $a>b$ $\nu = 0{\cdot}3$	$\sigma_C = \sigma_P - \frac{\sigma^2_P}{4\sigma_P}$ for $\sigma_R \leq 2\sigma_P$ $\sigma_C = \sigma_P$ for $\sigma_R > 2\sigma_P$	$\sigma_R = 0{\cdot}3E\frac{s}{r}$ Initial buckles taken into account
SHEAR	$\tau_P = k''_P \frac{E}{1-\nu^2}\left(\frac{s}{b}\right)^2$ for fixed edges $k''_P = 7{\cdot}5$ for supported edges $k''_P = 4{\cdot}4$ if $a>b$ $\nu = 0{\cdot}3$	$\tau_O = 1{\cdot}25\frac{Es}{b}\sqrt{\frac{s}{r}}$ for $\frac{b}{s}\sqrt{\frac{s}{r}} < 4{\cdot}3$, Initial buckles taken into account For $\frac{b}{s}\sqrt{\frac{s}{r}} > 4{\cdot}3$	$\tau_R = 0{\cdot}1E\frac{s}{r} - 75{\cdot}5E\left(\frac{s}{l}\right)^2$ for $l/r < 5$, Initial buckles taken into account
COMBINED COMPRESSION AND SHEAR	$\frac{\sigma_k}{\sigma_P} = 1 - \left(\frac{\tau_k}{\tau_P}\right)^2$	$\frac{\sigma_k}{\sigma_O} = 1 - \left(\frac{\tau_k}{\tau_O}\right)^2$ Not applicable to combined	$\frac{\sigma_k}{\sigma_R} = 1 - \left(\frac{\tau_k}{\tau_R}\right)^2$ tension and shear

Fig. 10
Buckling of the web of a plate girder, due to shear. (Note the diagonal tension field waves.

pression of a flat plate of the same width (See Table I.)

If a plate is under shear loads, e.g., in a plate girder, folds appear in the plate as soon as the buckling stress is exceeded (Fig. 10). To visualise the two-dimensional state of stress subsequent to buckling we have to consider the stresses prior to buckling. Fig. 11 shows that the direction of principal stress is 45° inclined to the direction of shear, and the principal stresses are $\sigma_1 = +\tau_{xy}$ and $\sigma_2 = -\tau_{xy}$. Mohr's stress circle, therefore, has its centre in the origin of the co-ordinate system. After buckling has occurred, evidently the greatest tensile stress is approximately in the direction of the folds (Fig. 10). Perpendicular to the folds, compression of only a limited magnitude can be carried. Thus, subsequent to buckling $|\sigma_1| > |\sigma_2|$. Hereby the centre of Mohr's stress circle shifts to the right out of the origin (Fig. 12), and the areas in the direction of the shear obtain direct stresses too. These direct stresses σ_x and σ_y (Fig. 12) must be transmitted from the plate edges to the reinforcing system. The state of stress between plate and edge reinforcements is shown in Fig. 13. The stresses σ_L and σ_Q in the longitudinal and lateral reinforcements result from internal equilibrium. We denote by s the sheet thickness, by F_L and b the cross sectional area and spacing of the longitudinal reinforcements, and by F_Q and a the area and distance of the lateral reinforcements. Then,

$$\sigma_x \, b s + \sigma_L F_L = 0 \text{ and } \sigma_y \, a s + \sigma_Q F_Q = 0,$$

and thus

$$\sigma_L = -\delta \sigma_x, \quad \sigma_Q = -\gamma \sigma_y \quad .. \qquad .. \quad (3)$$

if for brevity we put $b s / F_L = \delta$ and $a s / F_Q = \gamma$.

To calculate the stresses subsequent to buckling due to shear, H. Wagner has made assumptions which

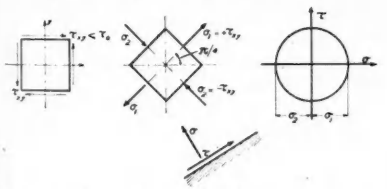


Fig. 11. Stresses, prior to buckling, in a plate under shear.

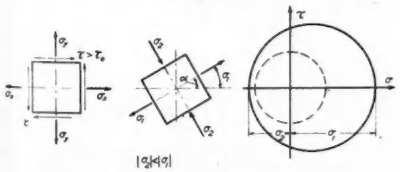


Fig. 12. Tension field stresses.

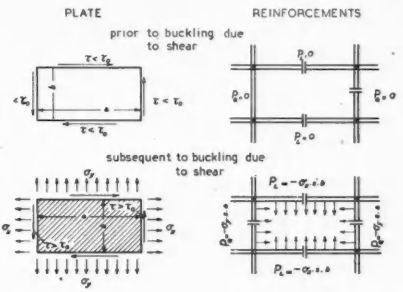


Fig. 13. State of stress between plate and reinforcements, subsequent to buckling due to shear.

enabled him to express the direction of principal tensile stress in a plate (Fig. 14).

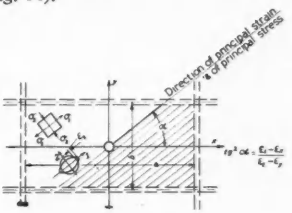


Fig. 14
Direction of stress and strain in a tension field.

This is given by $$\tan^2 \alpha = \frac{\epsilon_1 - \epsilon_x}{\epsilon_1 - \epsilon_y} \quad .. \qquad .. \quad (4)$$

Herein ϵ_1 is the principal tensile strain, ϵ_x and ϵ_y the component strains in the x and y direction.

To the definition of any two-dimensional stress three factors are required. Prior to buckling these are τ_{xy}, $\sigma_x = 0$ and $\sigma_y = 0$. Subsequent to buckling there is τ_{xy} and α, and in assuming a value for the principal compressive stress σ_2, the component stresses σ_x and σ_y as well as the principal stress σ_1 can be calculated:

$$\left.\begin{aligned} \sigma_x &= \tau \cot \alpha + \sigma_2 \\ \sigma_y &= \tau \tan \alpha + \sigma_2 \\ \sigma_1 &= \tau \tan \alpha + \tau \cot \alpha + \sigma_2 \end{aligned}\right\} \quad .. \quad (5)$$

According to what assumption can be made on the principal compressive stress σ_2, there are three different types of tension fields :—

1. The ideal tension field .. $\sigma_2 = 0$
2. The complete tension field .. $\sigma_2 = -\tau_0$
3. The incomplete tension field .. $\sigma_2 = f(\tau)$

The ideal tension field ("Wagner" field) offers the advantage of simple calculation; however, at low values of τ/τ_0 the stiffener stresses are overestimated. The latter drawback is not present in the complete tension field theory. A still better agreement with test results can be achieved by the incomplete tension field; but here, and also with the complete tension field, the calculations are more complicated than with the ideal tension field. In Fig. 15 these three theories are compared in representing them by Mohr's stress circles.

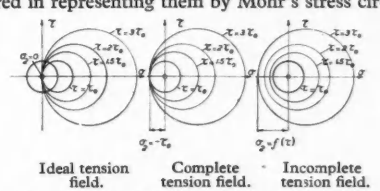
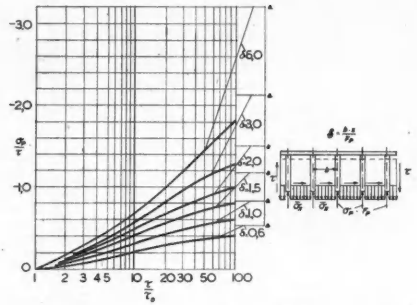


Fig. 15. Tension field theories represented by Mohr's circles.



a Asymptotes for $\tau/\tau_o = \infty$, where $\frac{\sigma_P^*}{\tau} = \delta \cot \alpha$

Fig. 16. Stiffener stresses in a plate girder.

After having assumed σ_2 in equation (5), α only remained unknown. It can be calculated according to equation (4) by substituting the following strains ($\nu = 0.3$) :—

$$\epsilon_1 = (\sigma_1 - \nu\,\sigma_2)/E, \quad \epsilon_x = \sigma_L/E, \quad \epsilon_y = \sigma_Q/E \quad .. \quad (6)$$

Here, σ_L and σ_Q is obtained from equation (3), and when substituted in equation (5) an equation of the 4th degree can be obtained and solved for α.

To what extent the incompleteness of the tension field does influence the secondary stresses can be shown by the example of a plate girder, the web of which in aircraft is made so thin that it buckles much below the "safe load." The stiffeners obtain compressive stresses σ_P, which for the ideal tension field can be calculated to $\sigma_P^* = -\delta\tau \cot \alpha$, where $\delta = bs/F_P$ and α counted from an axis perpendicular to the beam axis. The graph in Fig. 16 is based on test results of Wagner and Lahde, and shows that even at $\tau/\tau_o = 100$, i.e., the shear stress equals a hundred times that at buckling, the ideal tension field is not accomplished and the stiffener stress being a fraction only of the value $\sigma_P^* = -\delta\tau \cot \alpha$.

Elementary Calculation of Stiffened Shells.

The two basic assumptions of elementary shell calculation refer to bending and torsion respectively.

In bending, the assumption is that plane sections remain plane ; herewith, in the elastic range, linearly distributed stresses :—

$$\sigma = \frac{M_{by}}{I_y} z + \frac{M_{bz}}{I_z} y \quad .. \quad .. \quad (7)$$

M is the bending moment, I moment of inertia, y and z are the principal axes of inertia.

For torsion the assumption is constant shear flow t around the circumference, in accordance with Bredt's theory

$$t = M_D/2\,F_u \quad .. \quad .. \quad .. \quad (8)$$

Herein, M_D is the applied torque, F_u is the cross-sectional area enclosed by the middle surface of the skin plating (shaded in Fig. 17). Bredt's theory is valid only when the torque does not vary along the length of the shell, and the end section is not restricted against warping.

In the elementary calculation of a fuselage (Fig. 17) the direct stresses can be estimated by equation (7) unless buckling of the skin or diffusion of concentrated loads invalidate the assumption.

The total shear flow t is composed of bending shear and torsional shear :—

$$t = \frac{Q_z\,S_y}{I_y} + \frac{Q_y\,S_z}{I_z} + \frac{M_D}{2\,F_u} \quad .. \quad (9)$$

Herein S_y and S_z are static moments,

$$\left.\begin{aligned} S_y &= -\int_0^U z\,s\,du - \sum_1^K F_{Li}\,z_{Li}\,; \\ S_z &= -\int_0^U y\,s\,du - \sum_1^K F_{Li}\,y_{Li}\,; \end{aligned}\right\} \quad .. \quad .. \quad (10)$$

See Fig. 17 for the zero point ($u = 0$) which can be chosen arbitrarily ; F_{Li} is the cross-sectional area of the stringer at i, y_{Li}, z_{Li} the co-ordinates of the stringer centroid. Equation (10) is, however, valid for doubly symmetrical sections only ($u = 0$ being then at the intersections of the symmetry axis with the shell contour line). For shell in Fig. 17 (symmetry about one axis lacking) the correct values are : $S_y = S_{yo} + S_y(u)$ and $S_z = S_{zo} + S_z(u)$ .. .. (11) whereby $S_y(u)$ and $S_z(u)$ identical to the values given by equation (10). The initial values S_{yo} and S_{zo} can be calculated by referring to the condition of no angle of twist due to bending shear :—

$$\phi \frac{t}{s\,G}\,du = 0 \quad .. \quad .. \quad (12)^*$$

Herein,

$$t = \frac{Q_z}{I_y}\left[S_y(u) + S_{yo}\right] \text{ and } t = \frac{Q_y}{I_z}\left[S_z(u) + S_{zo}\right],$$

and it follows

* The integral ϕ is to be taken around the whole shell circumference.

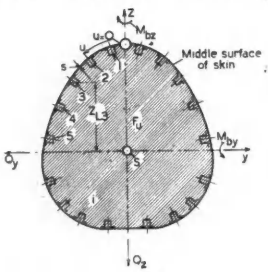


Fig. 17. Cross-section of a stressed skin fuselage.

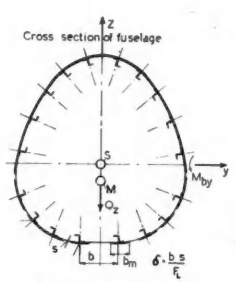


Fig. 18. Bending stresses, subsequent to buckling, in a stressed skin fuselage.

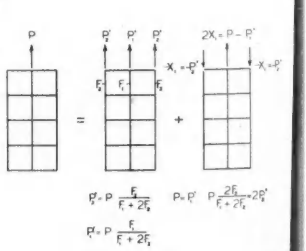


Fig. 19. Diffusion of a concentrated load into a panel with three longitudinal members.

$$S_{yo} = -\frac{\phi \frac{S_y(u)}{sG}}{\phi \frac{du}{sG}} \text{ and } S_{zo} = -\frac{\phi \frac{S_z(u)}{sG} du}{\phi \frac{du}{sG}} \quad .. (13)$$

To calculate the bending strength of a shell fuselage, the lessened load-carrying capacity, subsequent to buckling of the plates on the compression side, must be taken into account. It seems advisable first to calculate the direct stress σ as if the skin plating would carry the full load, and then in accordance with the effective width the actual stress σ_L in the longitudinals (stringers). This is shown in Fig. 18 :—

$\sigma(bs + F_L) = \sigma_L(b_m s + F_L)$ [compare with eq. (1)]

and thus :—

$$\sigma_L = \sigma \frac{1+\delta}{1+\frac{b_m}{b}\delta} \quad .. \quad .. (14)$$

with $\delta = bs/F_L$. This stress increase concerns the compression side only, and as b_m/b is not constant over the cross-section and therefore the distribution of σ_L is not linear (Fig. 18).

The treatment of wings and similar bodies is more difficult as the principal axes of inertia are not directly known. The subject is dealt with by Hertel (*L.F.F.*, 1931, pp. 1-56) and Ebner (*L.F.F.*, 1937, pp. 179-190).

Diffusion Problems.

The changes in the elementary stress distribution (Figs. 3-5) can be simply explained by considering the elementary loading, and superimposed on it a group of loads in equilibrium. Examples are given in Fig. 19 and Figs. 20 and 21. In the first example the group of loads is symmetrical with respect to the middle longitudinal of the panel; in the second the group is asymmetrical, i.e., the four warping loads form two moments of equal magnitude and opposite sign. The transfer of load P from the middle longitudinal (Fig. 19) into those on both sides corresponds to a dying away (decay) of the equilibrium group. Similarly, in Fig. 20, the non-elementary shear distribution at the built-in end changes into that due to the Bredt theorem, whilst simultaneously (Fig. 21) the group of loads and their shear flows decay to zero.

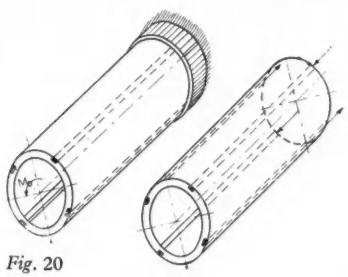


Fig. 20

Fig. 21

Torsion of cylindrical shell restricted against warping (Fig. 20) = elementary torsion + warping loads (Fig. 21).

The longitudinal forces present in a box beam under torsion, restricted against warping, be concentrated in the longitudinals (flanges), whilst the walls transmit shear forces only (Figs. 22 and 23). In the case of torsion with free warping the flanges are free of forces. To calculate the warping angle we compare the angle

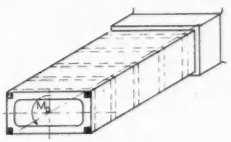


Fig. 22. Torsion restricted against warping.

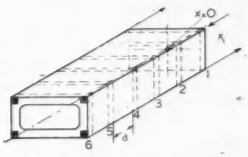


Fig. 23. Group of warping forces at restricted warping.

of twist ϑ with the shear strain τ/G. The warping angle in conformity with Fig. 24 is then

At the width $\varphi b = \vartheta\, c/2 - t/s_b G_b$.

At the depth $\varphi c = \vartheta\, b/2 - t/s_c G_c$;

herein $t/s_b G_b$ and $t/s_c G_c$ are the shear stiffnesses at constant shear flow t. Substituting from Bredt's theorem

$$\vartheta = \frac{M_D}{4b^2c^2}\left(\frac{2b}{s_b G_b} + \frac{2c}{s_c G_c}\right) \text{ and } t = \frac{M_D}{2\,bc}$$

the free warping distortion—defined as the displacement of one corner from the original plane—can be obtained (Fig. 24) as

$$w = \varphi_c(+c/2) = \varphi_b\, b/2 + \varphi_c\, c = \frac{M_D}{8\,bc}\left(\frac{b}{s_b G_b} - \frac{c}{s_c G_c}\right) \quad (15)$$

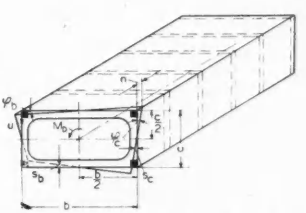


Fig. 24. Warping angle and free warping.

Now if one end of the beam is built in (Fig. 22), warping is restricted there. Four longitudinal forces X_1, being the group of forces which enforce a warping that cancels the free warping. The longitudinal forces die away along the beam length. Their momentary value

$$X = X_1\, e^{-\lambda(\xi-1)} \quad .. \quad .. (16)$$

wherein $\lambda = \cosh^{-1}(\alpha/2)$, $\alpha = 2 + \frac{16a^2(sG)\,L}{E\,F_L(b+c)}$ [index L denotes longitudinal walls (four)]. Further, by using X for the distance from the built-in end, and a for the length of one cell (Fig. 23), the co-ordinate employed is $\xi = X/a + 1$. Equation (16) is valid for the infinitely long beam. The beam is thought subdivided by lateral "walls" (frames) into "cells" which are taken here to be of equal length a. The ratio of X/X_1 is plotted in Fig. 25 against ξ for two types of frames. It has been shown in this example that diffusion problems cannot be dealt with simply by St. Venant's principle but require more exact treatment.

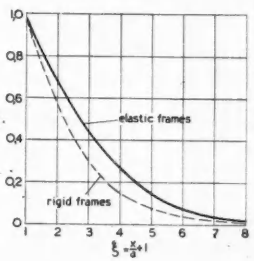


Fig. 25

Dying away of the equilibrium group.

THE COUPLING OF THE BLADES OF VARIABLE PITCH AIRSCREWS.

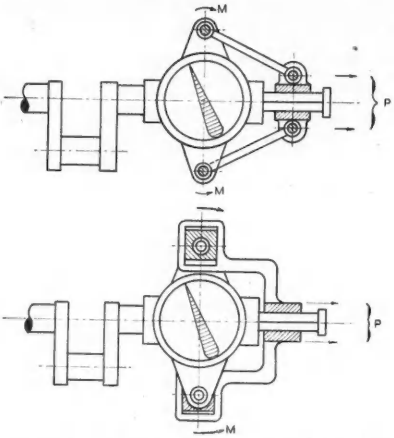
By Dr. Ing. W. Nitzsche, Kapfenberg/Steiermark. (From *Luftwissen*, Vol. 10, No. 3, March, 1943, pp. 74-77).

In this critical review all the possibilities of gearing are discussed which are open to affect the change of pitch of airscrew blades.

These pitch changing devices are usually referred to as coupling mechanisms, and they form part of all variable pitch airscrews, automatic or otherwise. Their task is to ensure equal and opposite setting of the blades, an operation which must be within close limits of accuracy as to the change of pitch, with the further requirement of smoothness and steadiness of action.

For the setting of the blades which are rotatable in the hub, a couple M must be applied to the blade root. If this couple is produced by a control couple, the coupling mechanism will be designated as an M-gear. If the couple on the blade root is produced by a control force P, the type of mechanism so described will be of the designation P-M. In either type of mechanism, a "linkage" on a gear, can be employed. The linkage may be a rigid or flexible link (belts and chains) or a "hydraulic link." The gear drive couplings can be sub-divided into pin, cam, screw, spur wheel, bevel gear, spiral gear and worm gear drives.

The figures below illustrate the design of each type of coupling mechanism. Coupling mechanism with rigid links are often provided for blade setting (Fig. 1-3) as they are simple in design. Their drawback



Figs. 1 and 2. Coupling mechanism of the rigid link type, P-M drive.

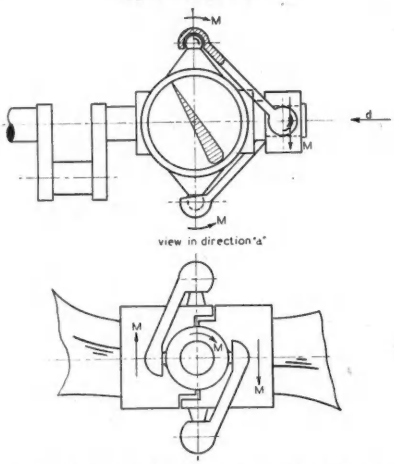


Fig. 3. Coupling mechanism of the rigid link type, M-M drive.

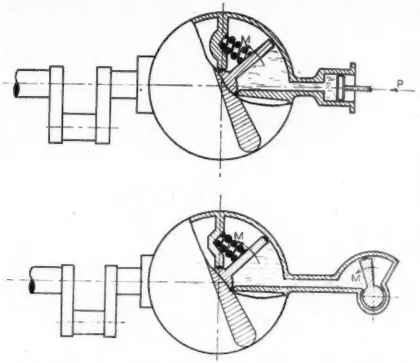


Fig. 4 (Top). Hydraulic coupling mechanism, P-M drive.

Fig. 5 (Bottom). Hydraulic coupling mechanism, M-M drive.

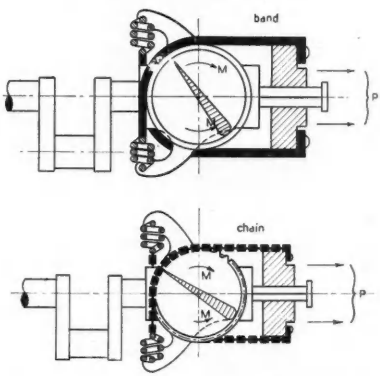


Fig. 6 (Top). Coupling mechanism of the flexible link (Belt) type, P-M drive.

Fig. 7 (Bottom). Coupling mechanism of the flexible link (Chain) type, P-M drive.

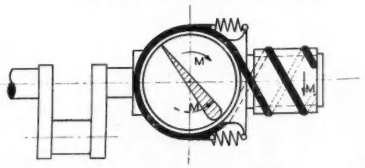


Fig. 8. Coupling mechanism of the flexible link (Belt) type, M-M drive.

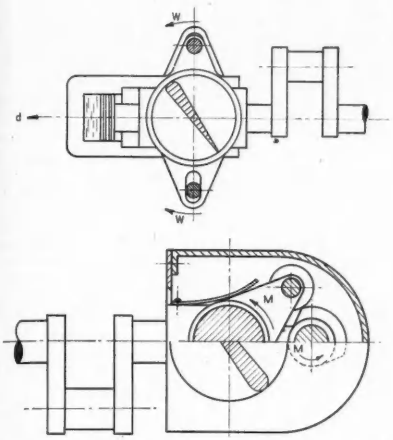


Fig. 9 (Top). Coupling mechanism employing pins, P-M drive.
Fig. 10 (Bottom). Coupling mechanism employing cam, M-M drive.

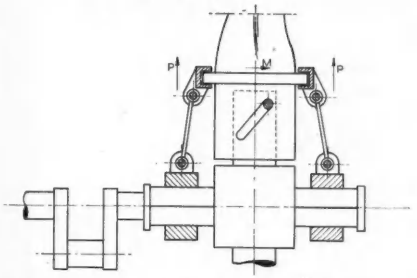


Fig. 11. Coupling mechanism employing pin and screw, P-M drive.

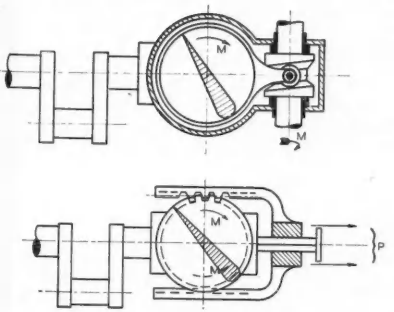


Fig. 12 (Top). Coupling mechanism employing single screw gear, M-M drive.
Fig. 13 (Bottom). Coupling mechanism of the spur gear type, P-M drive.

is that they are not self-locking, so that all alternating loads and vibrations of the blade are transmitted into the gear. The link itself is under alternating tension and compression, so that there is a danger of fatigue. The bearings of the links require large areas for the pins or balls to prevent rapid wear. With hydraulic links (Figs. 4-5) it is difficult to maintain a given blade setting (hunting of the controls at pitch changing), not to speak of the fact that to maintain a constant pressure, special control devices are necessary which take care of leakage and of the changes in liquid volume due to temperature variations.

Flexible links (Figs. 6-8) are seldom used in the way shown here, as with belts the phasing is difficult due to elongation and slip of the belt. Again, with chains the likelihood of breakage due to fatigue is prohibitive. The gear drive couplings offer in general an advantage of a fixed ratio between translation and rotation of the respective components.

Pin, cam, slide and single screw drives (Figs. 9-12) are used sometimes with success. Their drawback is that large control forces have to be transmitted by small arcs (line contact), resulting in high bearing pressures.

Toothed wheel gearing (Figs. 13-19) can be employed usefully by choosing a high gear ratio between control drive and blade, and so reducing the loads on the tooth system. This reduces wear and weight as well.

The bevel gear coupling is advantageous from the point of view of accuracy which is achieved by a single

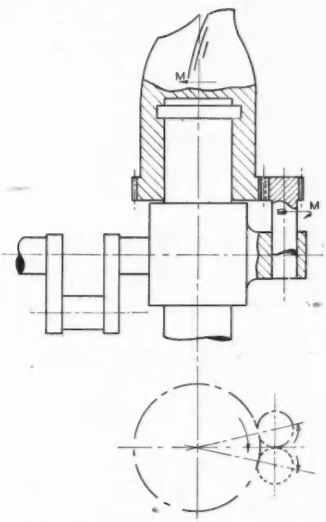


Fig. 14. Coupling mechanism of the spur gear type, M-M drive.

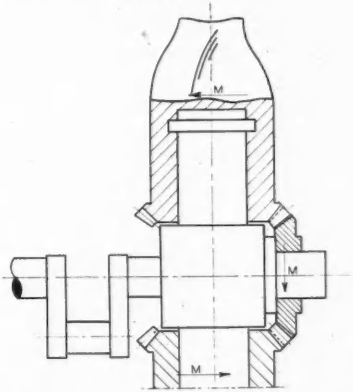


Fig. 15. Coupling mechanism of the bevel gear type, M-M drive.

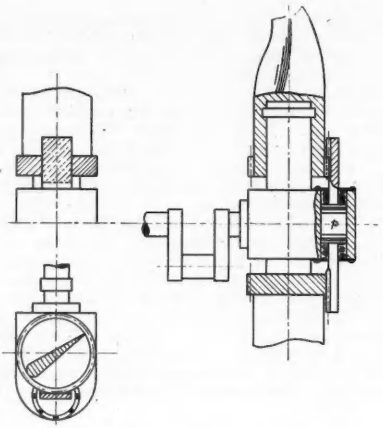


Fig. 16. Spiral gear coupling, P-M drive.

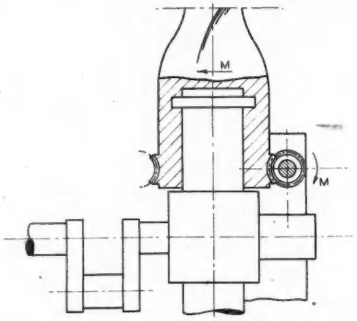


Fig. 17. Worm gear coupling, M-M drive.

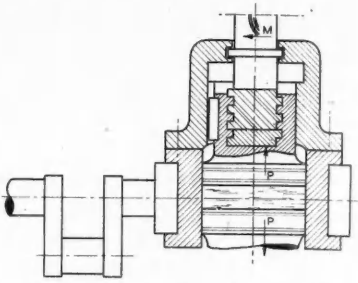


Fig. 18. Concentric screw gear coupling, P-M drive.

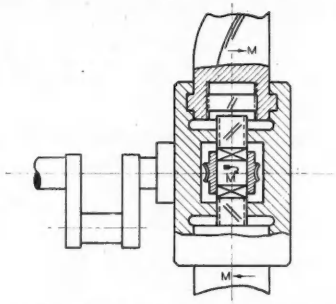


Fig. 19. Concentric screw gear coupling, M-M drive.

control wheel, however, the gear is not self-locking. With spiral gear, screw and worm gear drives (Figs. 16-19) self-locking is easily attainable. The suggestions in Figs. 16, 18 and 19 are of little practical value, and that leaves us with the worm gear Fig. 17 as the most favourable solution.

Note of the EDITOR, E. D.—*In the author's opinion the hydraulic coupling should be ruled out since there are many inherent disadvantages in its operation.*

SAVING MATERIAL AND LABOUR IN THE MASS PRODUCTION OF ELECTRIC APPARATUS.

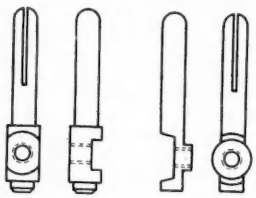
By H. SCHLAEGEL, VDE, Berlin. (From *Maschinenbau, Der Betrieb*, Vol. 22, No. 1, January, 1943, pp. 5-7.)

PRODUCTION under war-time conditions, with the consequent shortage of raw materials, skilled labour, and coupled with the necessity for increased output, has resulted in the introduction of new methods by designers and production men, and the principles underlying these methods will certainly lead to still further developments in the near future.

Use of Metal-Working Methods.

In mass production, costs are very largely influenced by the cost of raw materials. Economy in the use of raw materials is, therefore, very essential, hence the preference given during the past few years to drawing and squeezing operations instead of turning. One example is the introduction of cold-forged head screws and similar parts up to M12 (12 mm. metric thread) with rolled threads. For a hexagon-head DIN-standard screw the amount of material required has been reduced by more than 60%. With the development of automatic machines for rounding the screw ends, even for electrical contact screws, rolled types may now be used.

Another example is to be found in contact plugs (Fig. 1), which are made of steel instead of brass, the new type being made from round bars automatically cut to length and rounded off. For making the connection the lower part is cold-forged. A saving of 60% is effected in materials, while in labour the saving amounts to 30%. With metal cutting methods, where steel is



Turned & milled. Squeezed.
Fig. 1

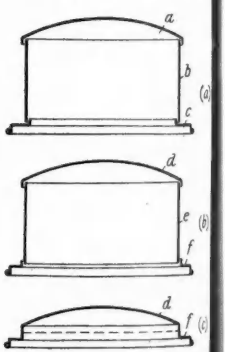


Fig. 2 (right) (*a*) Older model with cover *a* and ring *c* made separately; (*b*) New design with *d* and *f* made from one blank; (*c*) Cover *d* and base ring; (*f*) Stamped in one piece.

used for plugs instead of brass, double the time would be needed owing to the lower cutting speeds.

In other cases cold-working processes have saved at least 80% of the material needed for the original turned part. In such cases the automatic lathe departments, which constitute one of the bottlenecks of production, have been considerably relieved.

Modification of Design.

It is often found that simple changes in design lead to considerable economies. The housing, as shown in Fig. 2, was formerly designed in such a way that the inner part of ring *c* had to be scrapped. By making *d* and *f* from the same blank it was found that 20% of the material and 15% of the labour could be saved. With larger pieces, replacement of cast parts by welded designs is one of the most important methods of saving.

For mass production a complete change-over to new raw materials is very often the chief aim.

In moulded plastics small metal insets were used, e.g., nuts (Fig. 3). More frequently now they are omitted and threads are either moulded, or only the centre hole is moulded and the thread cut afterwards. This simplification is not possible, however, if the screw must be unscrewed and replaced frequently.

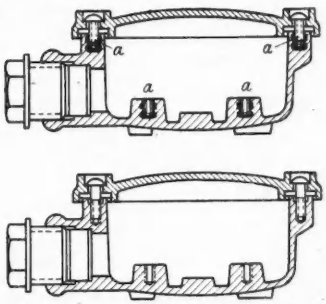


Fig. 3. Omitting brass insets in a moulded casing. (*a*) Brass nuts.

It is essential that the materials used for the change-over should, as far as possible, be produced in the country in which they are used. The counter wheels in Fig. 4 were formerly alloy die castings, while now they are composite aluminium stampings. Moreover, the new wheels weigh only 5 g. as compared with 27 g. for the tin wheels. They consequently require smaller bearings, producing less bearing friction. The frame of the counter, formerly of nickel-plated brass, is also made from aluminium.

Standardisation.

It was only to be expected that standardisation would play a great part in the saving of materials and labour. An example of this development is to be seen in the

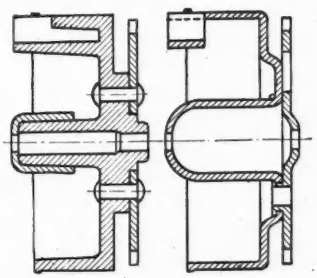


Fig. 4

(Left) Old model: Tin alloy die casting.

(Right) New model: Aluminium alloy, with short bearings.

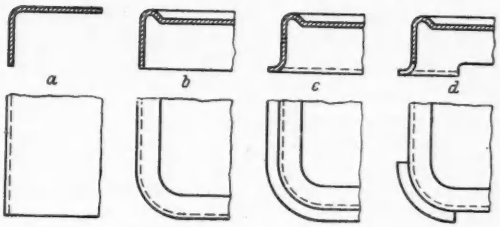


Fig. 5. Development of a base plate. (*a*) Cut off; (*b*) With a bulge on the edges; (*c*) With stamped edge; (*d*) Limitation of lower bulge to the corners enables using the same die for all sizes.

base plate for wall housings (Fig. 5). Model *a* was cut off straight and the housing cap had to be fixed by at least two screws. The next type, *b*, had a rounded edge and the cap was fixed with one screw only. The edges of type *c* were stamped instead of being cut off, thus effecting a considerable saving in time. Finally, type *d* is stamped off only on the four corners, as the four sides are kept shorter. Thus only one small die is needed for all sizes of the base plates.

Such modifications and changes also simplify the work of the designing department. Formerly, ceiling caps (Fig. 6) were chosen to suit the requirements of the manufacturers, while type *a* has now a neutral shape which is suited to all purposes.

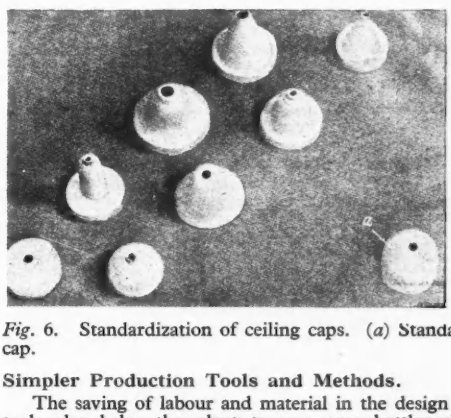


Fig. 6. Standardization of ceiling caps. (*a*) Standard cap.

Simpler Production Tools and Methods.

The saving of labour and material in the design of tools also helps the plant to overcome bottlenecks. Cap *a* (Fig. 6) is drawn in one operation with a single die. Formerly, a special set of tools was necessary for each type, as nearly all types were first spun and then finished by drawing. The new cap is not trimmed with cutters but stamped directly to size. The radius at the edge allows a reduction in thickness from 1 mm. to 0.6 mm. The total saving in material is 40% and the result in labour saving is up to 80%.

A very important method is cold extrusion, formerly used chiefly for tin, but now used for aluminium as well. This method, carried out with very simple tools, is advantageous for thick-bottomed parts which were formerly turned.

Hard brazing used to be an expensive and complicated method, used only where it was indispensable for strength or tightness. Other methods hitherto used, such as flame or dipping brazing, or using a brass bath, needed finishing operations to eliminate either scale or

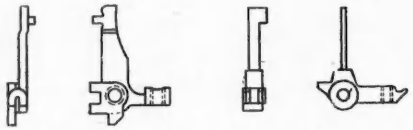


Fig. 7. Simplified manufacture of a lever. (Left) Original model; (Right) New model.

excess metal. Recently, brazing in a continuous furnace in a protective gas atmosphere has also been introduced in Germany.

A very important measure is the use of sub-assemblies and components for a whole range of products.

The designer can furnish decisive contributions towards simplifying tool production. A small lever (Fig. 7, left) was made from sheet metal in twenty-five operations. The re-designed part, right, is die-cast from zinc alloy and needs only five operations. Thus only five tools, which are simpler than those previously used, are needed.

TRANSMISSION OF HEAT BETWEEN VERTICAL WALLS AND TURBULENT WATER FILMS.

By U. GRIGULL VDI, Braunschweig. (From *VDI-Zeitschrift*, Vol. 86, No. 27-28, July, 1942, pp. 444-445.)

IF steam condenses on a cooled wall in the form of a continuous water film, the water starts flowing downwards in a laminar stream. In this case the transmission of heat is calculated according to Nusselt's Water Film Theory. As new condensate increases the thickness and mean velocity of the water film, the flow becomes turbulent if the wall is sufficiently high. It is possible to calculate the heat transmission for this case from Prandtl's theory of the analogy between the heat exchange and the exchange of impulses. The distribution of the water velocity is calculated according to the law of the 1/7 power and the shearing stress on the wall according to Blasius. The result, for the condensation of stationary saturated steam on a vertical wall, is given by

$$Re_H = f(Pr, A) \quad .. \quad .. \quad (1)$$

in which equation:

$Re_H = \overline{w}\,\delta/\nu$ Reynolds number of the water film at the vertical distance H(m) from the upper edge of the wall.

$\overline{w}$ (m/sec) the mean velocity of the condensate.

δ =(m) the thickness of the water film,

$\nu = \eta/\rho$ (m²/sec) the kinematic viscosity,

η (kg. sec/m²) the dynamic viscosity,

$\rho = \gamma/g$ (kg. sec²/m⁴) the specific Mass,

γ (kg./m³) the specific weight,

g = 9,81 (m/sec²) the acceleration of gravity,

$Pr = \nu/a$ Prandtl's number,

$a = \lambda/c\gamma$ (m²/sec) the coefficient of heat transmission,

c (kcal/kg., °C) the specific heat,

$$A = \frac{\lambda\,(\vartheta_s - \vartheta_w)\,\gamma\,2/3\,H}{1{,}082\,r\,g\,4/3\,\eta\,5/3}$$ a dimensionless coefficient,

ϑ_s (°C) the temperature of the saturated steam,

ϑ_w (°C) the temperature of the wall,

r (kcal/kg.) the heat of the vaporization.

These values all refer to the condensate.

Nusselt's theory, in the form of equation (1) reads:

$$Re_H = A^{0{,}75} \quad .. \quad .. \quad (2)$$

Between Reynold's number Re_H and the mean heat transmission coefficient $\overline{\alpha}$ (kcal/m² sec. °C) for the laminar and the turbulent portion, there is the relation

$$Re_H = \frac{\overline{\alpha}\,(\vartheta_s - \vartheta_w)\,H}{\eta\,r\,g} \quad .. \quad .. \quad (3)$$

The quantity of condensate flowing down per unit width of wall is

$$G_H = \eta\,g\,Re_H \text{ (kg./sec.m)} \quad .. \quad (4)$$

Equations (3) and (4) are valid for any type of flow. The turbulent flow of the water film starts at a critical Reynolds number of $Re_{kr} = \overline{w}\,\delta/\nu$ which is presumably between 300 and 400. As a first approximation, R = 350.

Fig. (1) shows equation (1) and (2), and the result of experiences conducted by various authors. The fundamental agreement between tests and theory is obvious. Downward steam velocity of over 5 m/sec. increases the heat transmission. This occurs probably at the high values of Fig. 1. Re_H and $\overline{\alpha}$ increase, too, if part of the wall is covered with drops instead of with a continuous film. The theoretical values constitute a lower limit, on which the design of a steamheated heat exchanger can be based. Fig. 1 is useful for that purpose, as equation (1) is not very handy for numerical calculations. For an approximate value of Pr = 5, a straight line may be drawn in Fig. 1 according to

$$Re_H = 0.74 \cdot 10^{-2} \cdot A^{1.4} \quad .. \quad .. \quad (5)$$

It shows the heat transmission for condensation if $Re_H > 350$. For smaller values, we have equation (2)

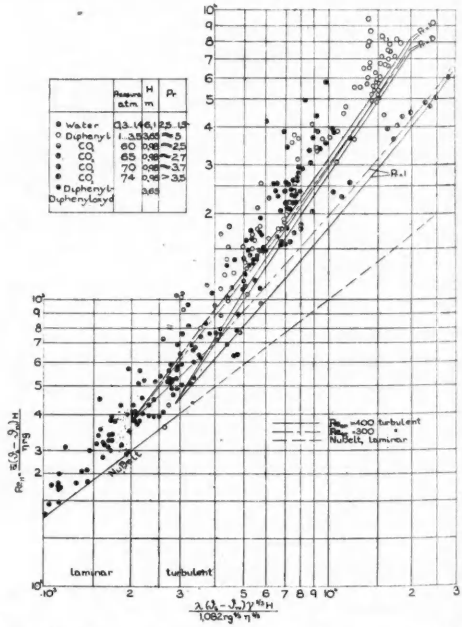


Fig. 1. Transmission of heat between vertical walls and turbulent and laminar water films. Dimensionless representation of Re_{kr} = 300 and 400.

for a laminar water film. With $Re_H = Re_{kr}$, we may calculate from equation (2) the height at which the flow becomes turbulent. Table I. shows for some substances the product H $(\vartheta_s - \vartheta_w)$, measured in (m C).

The coefficients characterizing the condensate are related to the temperature of saturated vapour ϑ_s, so that the values of the table may be somewhat modified depending upon $(\vartheta_s - \vartheta_w)$; e.g., for steam of 100° C and $(\vartheta_s - \vartheta_w) = 20°$ C, turbulence occurs after 2 to 3 m. Such distances actually occur in practical applications. According to Fig. 3, the heat transmission coefficient for turbulent film is higher as for laminar one and increases also with the distance over which the condensate flows, as opposed to the behaviour of a laminar film. From the point of view of heat exchange, it is therefore advantageous to design long vertical cooling paths, in order to make use of the advantages of the turbulent film. It is worth mentioning that evaporators which are very high for other reasons (e.g., Kestner-type ones), give better heat transmission also on the steam heating side. If heat exchangers with turbulent film are calculated according to Nusselt's theory, $\bar{\alpha}$ becomes too small and the apparatus may become too large.

TABLE I.

Length of the laminar cooling path for $Re_{kr} = 300$ and $= 400$. p_s = Vapour pressure, ϑ_s = Temperature of saturated steam.

Material	P_s at	ϑ_s °C	$x_o (\vartheta_s - \vartheta_w)$ m °C $Re_{kr} = 300$	$Re_{kr} = 400$
Water	0.1	45.5	164	240
	1.03	100	41.6	60.9
	100	310	6.53	9.58
Ammonia	4.38	0	29.5	43.3
	8.74	20	26.8	39.3
Diphenyl	1	255	19.8	29.2
Carbon dioxide	60	21.1	2.39	3.50
	65	24.6	1.84	2.69
	70	27.9	1.20	1.76

A NEW LOWERING BRAKE FOR THREE-PHASE ELECTRIC CRANES.

By KURT FLEISCHMANN. (From *Elektrotechnische Zeitscrift*, Vol. 63, No. 45-46, November, 1942, pp. 535-538.

ALL conveying machinery used for lowering heavy load must be equipped with efficient brakes. It is a sign of good control if the curve of speed plotted against the momentum of the descending load is flat, and if this effect is obtained by simple and secure means. The, Leonard system is the best one, but expensive, and therefore, a simple three-phase system is needed.

The reverse-current connection is the simplest method. The braking characteristics are steep, and therefore, unfavourable; but the current demand from the power supply is not larger than for hoisting the same load. Other braking systems destroy the symmetry of the phases, and thus the rotating field. Still another system uses a combination of the induction motor, driving in the sense of lowering the load, with a D.C. generator loaded by resistances. A last group uses D.C. to excite the whole or a part of the field windings. In all the latter systems the D.C. is separated from the A.C. mains, and the A.C. is prevented from penetrating into the D.C. circuits.

In the new system, the D.C. is superimposed on the whole of the normal, star-connected three-phase A.C. circuits. Such systems have been used for the control of machine tools and have proved that neither does the D.C. affect adversely the A.C. system, nor does the A.C. affect the D.C. generator in respect of commutation or in any other way. The only restriction is that either the D.C. generator must not have commutating poles, or the carcass and the commutating poles must be laminated. The D.C. generator is rated approximately for the rated current of the hoisting motor. Its rated voltage is equal to the voltage drop in the star-connected hoisting motor stator plus the drop in the A.C. supply system. In large nets this additional drop is practically zero, in unfavourable cases it is only a few per cent. of the drop in the stator. Fortunately such machines are of standard design, e.g., for electroplating.

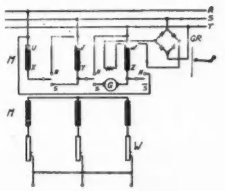


Fig. 1. Simplified diagram of connections of the new system. G, D.C. generator; GR, Rectifier; M, Hoisting motor: UX, VY, WZ, Windings of the motor stator; W, Resistances; Position H for hoisting; Position S for lowering.

Fig. 1 shows schematically the connections. The motor M is connected in mesh for lifting, in star for lowering. The saturation of the motor is decreased to 58% for lowering, because with normal saturation the D.C. would not create a field. The three-phase field rotates in the same sense as when hoisting; so we have, on the A.C. side, a reduced reverse current connection. The D.C. drives the motor M as a synchronous generator, discharging into the resistance W. Two currents pass through W—one of slip frequency sf, the other of frequency f-sf, which is proportional to the motor speed. The braking moment, therefore, consists of a normal asynchronous part M_a and a part generated by the motor acting as a synchronous generator M_g.

Fig. 2 shows this latter part M_g only, plotted against

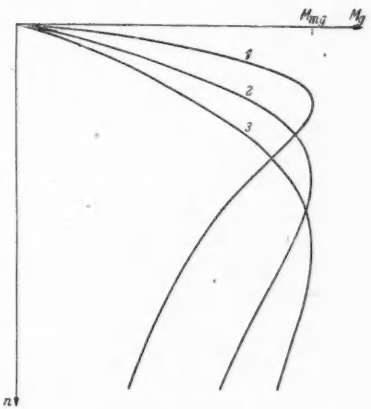


Fig. 2. Braking moment generated by the D.C. field, plotted against r.p.m. at constant D.C. excitation and for different resistances W.

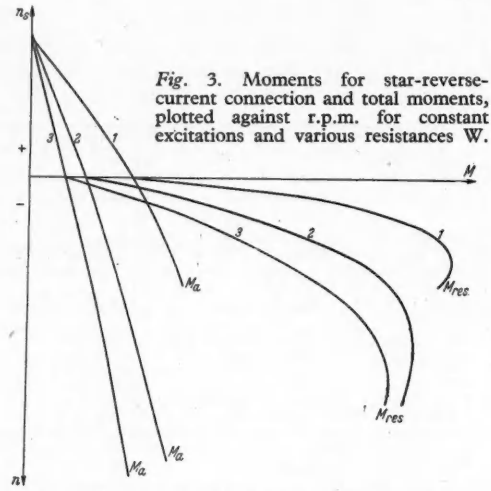


Fig. 3. Moments for star-reverse-current connection and total moments, plotted against r.p.m. for constant excitations and various resistances W.

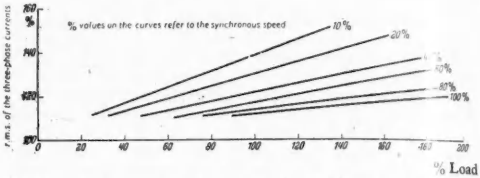


Fig. 4. Average stator currents for different loads and r.p.m.

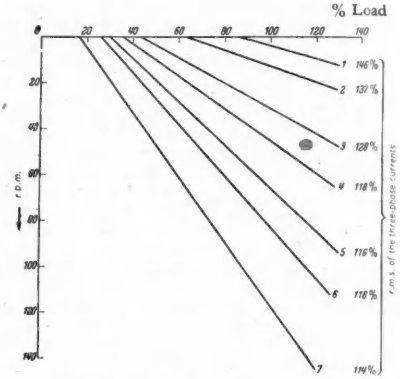


Fig. 5. Recorded braking characteristics with indications of the mean current intensity, for 100% superposition of D.C.

the r.p.m. Curve 1 corresponds to the smallest, curve 3 to the largest resistance, while the D.C. excitation is in all three cases the same.

Fig. 3 shows the curves of M_a for star-reverse-current connection and $M_{res} = M_a + M_g$, for the same condition. The resulting graphs are very flat, as is required. Their reversing points are far above full load.

The D.C. generator G (shown in Fig. 1 without its driving motor) is excited either by a dry rectifier GR or an exciter dynamo. Its power is 2-3%, that of the rectifier about 1% of that of the hoisting motor. As it is easy to have a very fine or smooth regulation of the excitation, and thus to turn the characteristics M_g, Fig. 2, around the origin of the co-ordinates, the characteristics M_{res}, Fig. 3, may sweep over large areas. The new system thus makes very fine regulation possible.

Fig. 4 shows the speed and the phase currents; the increase of current for braking is small and is less than for any other connection with the exception of those having a special braking generator, and of a pure reverse current connection.

The braking energy drawn from the power supply is small, as the motor is star-connected.

There must be also the possibility of driving when lowering. When switching over, free descent of the load must be avoided; this is done by changing the sense of the rotating field without interrupting the D.C.

Fig. 4 and 5 have been measured on an existing crane. The characteristics in Fig. 5 are straight ones; the curved parts corresponding to Fig. 3 being above 130% load.

SHEATHING OF ELECTRIC CONDUCTORS WITH POLYVINYL CHLORIDES.

By Hans Beck and Arnolf Rehbock VDI, Ludwigshafen o/Rhine *). (From *VDI-Zeitschrift*, Vol. 86, Nr. 41-42, 17th October, 1942, pp. 629-632).

Electric conductors are sheathed with rubber insulation either by rolling processes or by extrusion methods. The first are widely used in Germany; strips of the unvulcanized rubber are pressed on the conductor by profiled rollers. The sheathed wire is then vulcanized; the insulation has two longitudinal seams. The second method, used chiefly in the U.S.A., consists in extruding the unvulcanized rubber through a nozzle containing a hollow mandrel through which the conductor is passing (Fig. 1). The nozzle is usually a transversal one. The rubber is fed into the machine, caught by the feed worm (a) and pressed through the heated cylinder (b) in the nozzle (c). Here it flows round the mandrel (d), is deflected by 90° and in the mouthpiece (e) the sheathing is formed. Several wires may be worked in parallel by one machine.

Screw extrusion machines are in principle suitable materials having plastic properties, and are well suited for working thermo-plastic synthetics, but while rubber compositions are worked at moderate temperatures, synthetics need higher ones to become plastic. E.g., for polyvinyl chlorides (Igelit PCU) the temperatures are between 160° C. and 180° C. At lower temperatures, working is possible, but tensile strength and other properties deteriorate. Higher temperatures cause overheating or decomposition of the synthetic.

Screw extrusion machines working at 160° to 180° C.

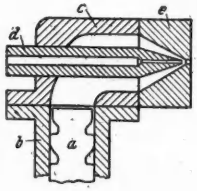


Fig. 1. Cross-section of a transversal nozzle for insulating electric conductors.

(*a*) Transporting screw; (*b*) Heated cylinder; (*c*) Nozzle; (*d*) Tube for guiding the wire; (*e*) Mouthpiece.

*28th Communication from the Plastics Department of I. G. Farbenindustrie A.G., Ludwigshafen.

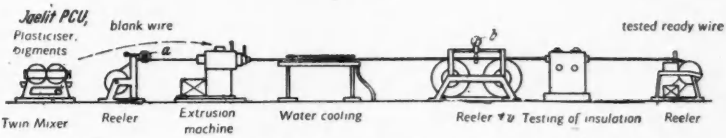


Fig. 2.

Sheathing plant for Plastics: (*a*) Speedometer; (*b*) Thickness indicator.

were not known hitherto and new types had to be developed. There is no final solution so far, as there have been several ways to attack the problem, but the new types are all characterized by using electric heating in preference to other methods, sometimes in addition to an older steam heating. Heating by hot oil is used less frequently. The heating system is mostly divided into zones to obtain a temperature rise on the way from the filling chute to the mouthpiece.

Feeding of Extrusion Machines.

According to the size of the plant and the existing machinery, different methods are possible; feeding with cold plastic, with warm plasticised one and finally with hot plastic. Cold or slightly heated plastic is fed into the machine in the form of flakes or ribbon, and on the way from feeding chute to mouthpiece it is homogenized, mixed and brought to at least 160° C. Because of the not very good conditions of heat transfer in the machine, the surface temperature of the heated walls must be high, otherwise the throughput would be too small. Thus the synthetic is easily damaged by overheating; if the temperature is kept low, cold particles may be dragged away and included in the sheathing. In both cases, faults in the insulation are the result. An advantage of this method is the absence of preparatory plant, such as kneading machines or mixing roller stands, as synthetics ready for extrusion are available. Output is rather low, thus an installation of this type is suited for smaller manufacturers. With an installation consisting of a dry-oven, a screw extrusion machine of 60 mm. screw dia. and the necessary feeding and winding reels, 2.5 sq. mm. cable can be produced at rates up to 20 m. per minute.

Output of an extrusion machine may be increased by feeding it with hot plastic. A simple arrangement to that effect consists of a pre-heating extrusion machine which warms, kneads and plasticises the raw material. The extruded strand is fed into the principal machine. With a preliminary machine of 100 mm. worm dia., a sheathing machine of 60 mm. dia. will be able to produce at 70 m. min., if the first machine is fed with ribbon or flakes. The first machine may be sturdy and simple, as there is no need for accuracy.

As the two machines have different tasks to fulfil, they must be of different construction, in order to attain high output. Design of the screws, number of threads, pitch, progression (= variation of pitch and thread depth along the screw), ratio of dia. to length and speed are essentially different in the two cases.

The most economic method of production is to preheat the plastic on heated mixing rollers which bring it to the full working temperature of 160° C. In that case polyvinyl chloride powder, mixed with plasticiser and pigments may be fed directly on the rollers. Two pairs of rollers are necessary, as preparation of a homogenous raw material takes a certain time. The two pairs of rollers are either used alternatively, or one pair is used for preparatory mixing and one for finishing. With 60 mm. screw dia., a 2.5 mm² conductor, twin roller stands with rollers of 300 mm. dia. and 650 mm. length each, an output of 120 m/min. is attained, corresponding to 1 kg of plastic per minute. With a necessary mixing time of 8 minutes this corresponds to feeding the rollers with 10 kg of compound, a quantity which can just be worked by rollers of the described size. The extrusion machine can handle this quantity, provided the screw is of adequate design, r.p.m. are sufficiently high and the nozzle is of modern construction. Fig. 2 illustrates a complete sheathing installation.

Design of the Nozzles.

The design of the nozzles determinates the output of the machine. Apart from the length of the nozzle, there are friction losses, deflection of the flow and the influence of any obstructions which reduce the throughput. Length of the nozzle cannot be reduced, as otherwise there would be no space for the hollow mandrel guiding the wire. The friction losses may be reduced by polishing and hard-chromium plating the parts concerned. Deflection of the flow through 90° in the usual transverse nozzles has been reduced to 60° and even 35° in modern oblique nozzles. A considerable resistance is generated by the mandrel. As it is usually provided with electric heating, a certain diameter is necessary. The plastic conveyed by the screw is under high pressure and will take the shortest path and, therefore, not have the same velocity at all points of the cross-section. At dead corners, the plastic flows slowly or may even stop, in which case it would eventually be burned. This effect is countered by providing an overflow opening at the point where the plastic is moving most slowly; in older machines without stepless regulation of wire velocity the overflow is also used to regulate the sheathing thickness. As a consequence of irregular distribution of velocity, the flow is not concentric and symmetric with the mouthpiece; there may appear a lateral thrust upon the wire, and the insulation may become excentric, too. Most screw extrusion machines have centering devices, by which the nozzle opening is adjusted relatively to the wire, but this may be done only with the machine running and control of concentricity is extremely difficult.

Development of a New Type of Nozzle.

These two drawbacks, burning of the plastic and excentricity, reduce production. They may be avoided by making the axis of the screw and of the wire coincide, i.e., by running the wire through the hollow screw. Most screws are provided with a hole to accomodate a cooling or a heating system; the heating system is unnecessary with the use of hot polyvinyl chloride or other thermo-plastics, and the hole becomes available for the wire.

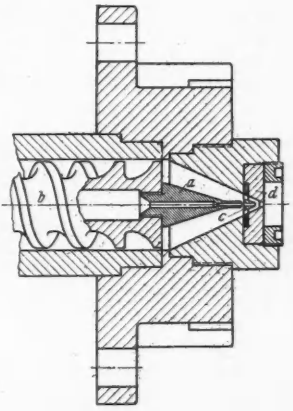


Fig. 3.

Cross-section of the new axial nozzle. (*a*) Tube for guiding the wire; (*b*) Transporting screw; (*c*) Centering ring; (*d*) mouthpiece.

In earlier designs of this type the wire was conducted through a stationary tube. These constructions all failed because of the difficulty of effectively sealing the joint between the rotating screw and the stationary tube against the plastic which is under high pressure. Long labyrinth seals, stuffing boxes, graphite and carbon seals, as well as bearings with hardened surfaces, all failed after working for a time because of excessive wear. The solution was found in using a rotating tube (Fig. 3) forming an extension to the screw. Because of the floating arrangement of the screw, it may not be used for centering. For that purpose, the wire is conducted through a small centering ring (c). The hole in the end of the rotating tube is larger by some tenth of mm. than the wire. The latter passes for a short distance freely through the plastic and, therefore, the force necessary to draw the wire along is greater than with the older designs. But the tension remains within limits admissible for the wire.

To obtain high output, it was necessary to use a screw with very deep threads, and screw speed was increased to 90 r.p.m. The plastic passes through the machine in only 30 sec. and, therefore, its intense heating is not so important. If the plastic is delivered by the rollers at 170° C., it is only cooled down slightly while passing through the machine and would certainly not fall below the minimum working temperature of 160° C. To obtain a bright surface finish, it would be advisable to heat the mouthpiece to 185° C.

As a consequence of the reduced heating of the machine—during the experiments, with steam at 47 lbs. p.o.i.—the plastic cannot be damaged during any temporary standstill of the extrusion machine. It will slowly cool down to the temperature of the machine. When starting afresh, this cooler mass must be drained; this concerns only some 500 gr., which can be re-used as it will not have been overheated.

Use of a feeding roller in the filling chute, turning in the opposite direction to the screw, greatly helps in feeding the machine and makes it unnecessary to force the material down by hand.

The advantages of the new extrusion nozzle are not limited to thermo-plastics, but may be extended to natural rubber and to Buna. Not only is output increased, but there are new outlooks for the technique of mixing.

Design of the Screws.

The screw used in the extrusion nozzle is purely a conveying screw of constant pitch. Screws with pitch towards the nozzle (progressive screws) exert a kneading action, but reduce output. They are used in machines fed with cold or moderately warm polyvinyl chloride, as well as in preparatory extrusion machines, but progression of the screw must not be exaggerated, as otherwise output is too small.

HOW TO MAINTAIN INDUSTRIAL TRUCKS.

By F. L. SAHLMANN. (From *General Electric Review*, Vol. 46, No. 6, June, 1943, pp. 325-328.)

REGULAR inspection and overhauling of the battery-powered industrial truck are the only positive guarantees for its continuous, efficient performance. The following suggestions are designed to facilitate the maintenance of the electrical parts of this important piece of equipment.

General Precautions.

The batteries, sole source of propulsive energy, should be serviced at regular intervals, according to the recommendations of the battery manufacturers. Both the traction motor, which propels the truck, and the hoist motor which drives the hoist and tilt mechanisms, should be inspected monthly and overhauled once a year. All the other electrical equipment should also be inspected and overhauled at like intervals. Suggested procedures are listed in the following paragraphs.

Monthly Inspection.

(1) Clean the commutator cover and surrounding parts to prevent dirt from falling into the motor.

(2) Remove commutator cover and examine mechanism. The copper surface should be smooth and free of grease.

(3) See that the brush holder mechanisms seat on brushes and that all terminals and connections are tight.

(4) Wipe carbon dust from cables and brush holders. Remove dirt from brushes by lifting springs, and raising and lowering brushes in the carbon ways. Do not snap the springs as this may chip the brushes.

(5) Replace short or broken brushes with new ones of the proper grade and length.

(6) Inspect the interior of motor for defect insulation or other injuries, and then replace damaged parts. Inspect connections for tightness.

(7) Replace commutator cover.

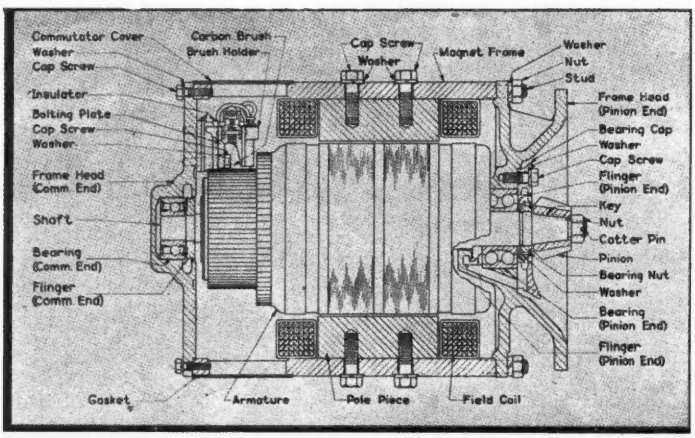


Fig. 1. Sectional view of a traction motor for a battery powered industrial truck.

Annual Overhauling.

(1) Remove motor from truck and clean the outside of it.

(2) Remove commutator cover and raise brushes from the commutator.

(3) Remove the nuts from the studs at the pinion end of the motor and withdraw the armature complete with commutator-end bearing assembly and pinion-end framehead assembly through pinion end. Keep bearings free of dirt.

(4) Remove the pinion-end framehead assembly and commutator-end bearing assembly from armature, employing suitable puller.

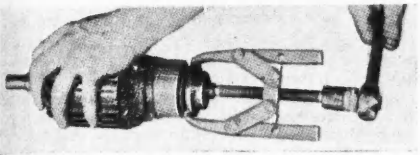


Fig. 2. Side view of armature showing front-end ball bearing being removed with hook-type puller.

(5) Blow out dirt from armature, using clean, dry compressed air and then wipe clean of oil and grease with a cloth saturated with carbon tetrachloride.

(6) If armature is found in good condition, proceed as follows: Bake at least 12 hr. at 250° F. While hot paint with a varnish such as Glyptal No. 1201. Keep the commutator clean. Bake for 12 hr. at 250° F. Note.—If armature coils are damaged, send complete armature to nearest service shop for repair.

(7) Take out the cap screws at the commutator end and remove the framehead.

(8) Blow the interior of the motor with compressed air and proceed as in (5).

(9) If field coils are in good condition paint them with varnish such as Glyptal No. 1201. Also, paint the interior of the motor, but keep pole-piece faces clean. Otherwise, remove field coils and poles, but keep each pole, coil, and any accompanying shims together. Upon reassembly, parts should be returned to their original position. Use new lock washers, clean contact surfaces, check coil polarity. Guard against loose connections.

(13) Clean brush holders and cables using carbon tetrachloride.

(14) Note that brush-holder mechanisms operate properly, that brushes are free in carbon ways and of same length. New brushes may be fitted to the commutator by placing a strip of fine sandpaper between the end of the brush and the commutator with the rough end of the sandpaper against the brush. Shunts, terminals and pole-piece cap screws must be tight.

(15) Reassemble parts on armature. When replacing bearings, use a suitable brass sleeve so that the pressure will be on the inner race of the bearings.

(16) Reassemble armature in frame.

Commutator.

The commutator should never be lubricated. Dirt and grease will cause short circuits. If the commutator should become rough, the armature should be placed in a lathe and the copper turned down just enough to give a true surface. The mica between the copper commutator segments is undercut 3/64 in. below the copper brush surface and should be undercut as often as necessary to prevent the mica from becoming flush with the copper brush surface. There should be no sharp edges on the copper segments. While reconditioning the commutator, prevent copper dust from lodging in the winding by using a suitable covering.

Hoist or Tilt Motor.

The same procedure suggested for the traction motor should be followed if the truck is equipped with a hoist or tilt motor, except for the differences in the method by which the motor is removed or opened.

Controller.

(1) Blow out dust with clean, dry compressed air.

(2) If required, oil contact cam rollers and reverser-shaft bearing. Use an accepted lubricant in a long-snout oil can.

(3) Replace any broken parts and check all connections for cleanliness and tightness.

(4) Remove any roughness on contact surfaces with a fine file and replace contact tips worn half-way through.

(5) At yearly overhaul period, remove, clean and repack main shaft bearings.

(6) Both the contact tip pressure and the distance between the tips when the contactor is open should be checked periodically.

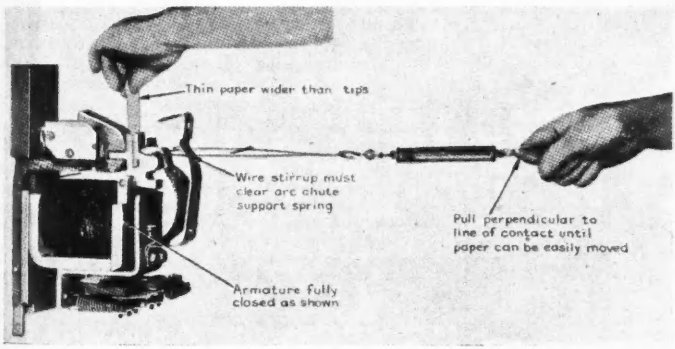

Thin paper wider than tips
Wire stirrup must clear arc chute support spring
Pull perpendicular to line of contact until paper can be easily moved
Armature fully closed as shown

Fig. 3. Checking the contact-tip pressure on a magnetic contactor.

Contactor.

(1) Blow out dust and grit with compressed air.

(2) Replace any worn or broken parts.

(3) Proceed as in (3), (4) and (6) for the controller. Note.—Specific information relative to coil operation and contactor adjustment should be obtained from the manufacturer.

Accelerating Resistor.

(1) Proceed as in (1) and (3) for the controller.

(2) Check both resistor ribbon and porcelain insulator for damage.

RIVET WIRE AND RIVETS OF Al-Cu-Mg ALLOY

By Ing. Sven Tobert. (From *Teknisk Tidskrift*, Vol. 73, No. 7, 10th July, 1943, pp. 53-58).

Rivets used in aircraft construction are usually of Al-Cu-Mg alloy with approximately 4 per cent Cu., 0.6 per cent Mg., 0.5 per cent Mn., 0.3 per cent Si., and 0.3-0.4 per cent Fe. Commonly, rivets are driven cold, a few hours subsequent to a solution heat treatment at about 500 °C. followed by rapid cooling. As the rivets are subject to rapid age-hardening at ordinary room temperatures, they must be driven within a short time after treatment in order to avoid the formation of cracks. Keeping the treated rivets on ice makes it possible to retain them in soft state for a somewhat longer time; but, in the author's opinion, this method cannot be considered practical; nor does the comparatively small gain in time justify its use. In actual shop practice, therefore, solution heat treatment is carried out in batches commensurate with the expected rate of demand.

To define the time interval during which solution treated rivets can be used, solely as a function of ageing time in a given temperature range, would be insufficient insofar as the effect of the original structure previous to solution treatment must also be considered. In this respect the degree of cold-working effected by the wire drawing process must be considered largely decisive. In the manufacture of cold drawn rivet wire, the common practice is to effect a 20-50 per cent reduction after the last anneal. This report present the outcome of an investigation made for the purpose of gaining further information with regard to the influence of the magnitude of reduction effected by cold drawing after the last anneal ; and the influence of repeated solution treatment upon the latter was also studied.

For the purpose of this investigation, a coil of 5.96 mm. dia. wire was divided into equal portion A and B. This was supplemented by another equal amount C, made up by wire of a somewhat different composition. These three samp.es yielded the following analysis :

These samples were treated as follows :

Sample	Cu. %	Mg. %	Mn. %	Si. %	Fe. %	Zn. %	Sn. %	Pb. %	Ti. %
A	4.0	0.50	0.51	0.28	0.34	0.05	0.01	0.01	0.01
B	4.0	0.56	0.51	0.28	0.34	0.05	0.01	0.01	0.01
C	4.0	0.57	0.50	0.25	0.30	0.05	0.01	0.01	0.01

A. Annealed at 5.96 mm. dia. and subsequently drawn to 4.5 mm. dia. equivalent to a reduction of 43 per cent.

B. Annealed at 5.96 mm. dia. and subsequently drawn to 4.95 mm. dia. ; annealed again and drawn to 4.50 mm. dia., equivalent to a reduction of 18 per cent.

C. Annealed at 4.96 mm. dia. and subsequently drawn to 4.50 mm. dia. equivalent to a reduction of 18 per cent.

From each sample about 600 rivets of 4.5 mm. dia. and 20 mm. length with round head were made. The remaining quantity of wire and the rivets were subjected to solution treatment at 500 °C. for 40 minutes. Strength values of the wire were determined 2, 4, 8, 16, 32, 64, and 128 hours after the solution treatment, the sheer strength of the rivets being ascertained at the same intervals

Test rivets were driven immediately and 4, 8, 24, 48, 72, 96 and 120 hours after solution treatment. In each of these tests, carried out with the use of a compressed air tool, 15 rivets were driven at the same time, joining two 8 mm. aluminium sheets.

Two days after the first solution heat treatment, this same treatment was repeated, and again repeated after a further two days. Both after the second and the third solution treatment, the wire samples and rivets were tested as before, and the identical numbers of rivets were driven. Fifteen days after the last test, four rivets were sawn off from each batch of rivets driven and were subjected to shear strength test.

All results of the strength tests were charted, each point plotted representing the mean value obtained from at least three separate tests. Fig. 1 shows the development of age-hardening at room temperature after one, two, and three solution treatments of wire samples A, B, and C. It is seen that after each individual solution treatment, the three samples showed identical trends of age-hardening. A comparison of the respective strength values found for the A and B samples consisting of the same material, showed that the A sample (obtained with 43 per cent reduction) has a maximum strength of about one kg. per sq. mm. higher than the B sample (obtained with 18 per cent reduction).

In contrast to this, the ultimate tensile and shearing strength values of the sample C wire, although drawn with 18 per cent reduction, approximate to the corresponding values obtained with the sample A wire. In the author's opinion, this may possibly be partly explained by the lower combined percentage of Si. and Fe. of sample C.

Fig. 2 gives the comparative effects of age-hardening with time after one, two, and three solution treatments respectively. These curves represent average values of samples of the A, B, and C varieties. It is seen that after the first solution treatment, and at the commencement of the ageing process, the wire samples are less "hard" than after the second and third treatments. But it should be noted that after some three to four hours, all three curves exhibit identical trends.

The influence of age-hardening upon shear strength of the rivets with one, two, and three solution treatments, is indicated in Fig. 3. This graph proves the influenc:

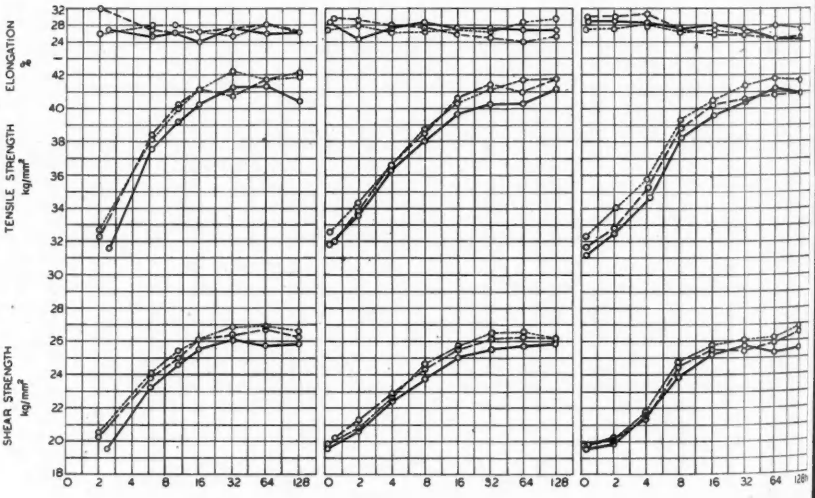


Fig. 1. — Age-hardening at 20 deg. C. after 1, 2, and 3 solution treatments.

..... Sample A drawn cold with 43 per cent reduction.

——— Sample B drawn cold with 18 per cent reduction.

- - - - Sample C drawn cold with 18 per cent reduction.

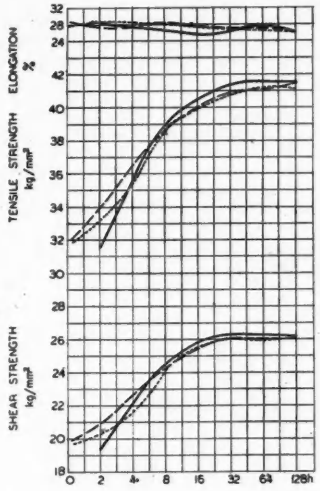


Fig. 2
Age-hardening *versus* number of solution treatments.
——— 1st solution treatment.
- - - - 2nd solution treatment.
..... 3rd solution treatment.

of the different methods of solution treatment to be rather insignificant.

The data derived from the rivetting tests proper are charted in Fig. 4, which is based on the A and C material. "Rivet failure" was defined as the appearance of cracks

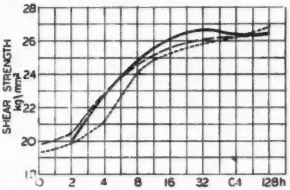


Fig. 3.—Shear strength of rivets *versus* time, with 1, 2, and 3 solution treatments.
——— 1st solution treatment.
- - - - 2nd solution treatment.
..... 3rd solution treatment.

in one of the rivet heads. The A variety of rivets in no case showed cracks in the original head; but a few cases of fissures in the driven heads were encountered. With the C rivets, however, the majority of cracks occurred in the original heads. This should be kept in mind when adjudging the failure statistics given in Fig. 4. The part of the graph based on one solution treatment further shows that under this condition the A and C varieties exhibit very similar respective failure characteristics. It is also seen that with one or two solution treatments, rivets of the C-variety cease to be usable after so short a time as eight hours; while the A-variety can still be used after 100 hours.

By far the greatest interest attaches to the influence of the time interval between treatment and use upon the shear strength of the driven rivets. In order to clarify this point, four rivets from each test run were removed 15 days after solution treatment and tested for shear strength. The results are shown in Fig. 5. Here the ageing process shows itself as influenced to a remarkable degree by the time interval between treatment and driving of the rivets. It follows that a rivetted joint made immediately upon solution treatment, may exhibit a shear strength 5-6 kg. per sq. mm. below that obtained with driving the rivets one day after treatment. Also, with rivetting immediately after treatment, the rivets

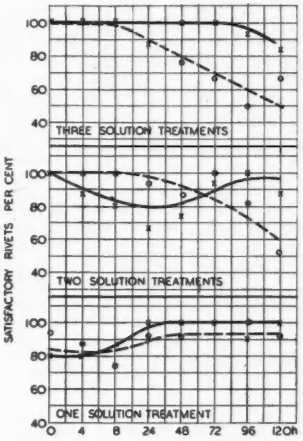

THREE SOLUTION TREATMENTS
TWO SOLUTION TREATMENTS
ONE SOLUTION TREATMENT
SATISFACTORY RIVETS PER CENT
0 4 8 24 48 72 96 120h

Fig. 4.
Rivet quality *versus* time of driving after solution treatment.
——— Rivets made from sample A.
- - - - Rivets made from sample C.

may not even show the specified strength of 25 kg. per sq. mm. (except rivets subjected to three solution treatments). But it is clearly seen that an interval of four hours between treatment and driving of the rivet will suffice to achieve specified shear strength. Another point of note in Fig. 5 is that in the range considered, the strength values of the A-variety are consistently one kg. per sq. mm. higher than those of the B and C varieties.

The microstructure of the A-wire drawn with 43 per cent reduction after the last anneal, is said to possess a double fibre structure, of which a small residue remains after solution treatment. With hardening, a fine-grained structure results. But cold-drawn wire made with an 18 per cent reduction shows no pronounced structure and becomes coarse-grained after solution treatment.

From these various observations, the following conclusions are drawn:

Rivets made from 18 per cent reduction material must be driven only four hours after solution treatment; and they should not be used later than 8 hours after treatment, if solution treatment was applied more than once. Rivets which have undergone only one solution treatment can be used for a somewhat longer time.

If made from wire drawn with 43 per cent reduction, rivets must be used not earlier than four hours after solution treatment; they become unusable after 4 to 5 days. The times given refer to ageing at 20 deg. C. and for materials with 0.5 per cent Mg. Falling Mg. content and lower ambient temperature increase the period for which they can be used, and vice versa.

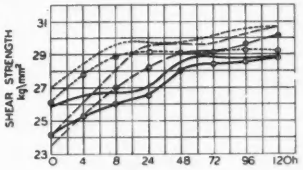


Fig. 5.
——— 1st solution treatment
- - - 2nd solution tretment } Sample A
..... 3rd solution treatment
-o-o- 1st solution treatment
o--o--o 2nd solution treatment } Samples B and C
o....o....o 3rd solution treatment

ARC WELDING OF MAGNESIUM AIRCRAFT STRUCTURES

By V. H. PAVLECKA and J. K. NORTHROP, Chief of Research and President, respectively, Northrop Aircraft, Inc., Hawthorne, Cal. (From *Metals and Alloys*, Vol. 18, No. 2, August, 1943, pp. 302-307).

DURING the last decade, monocoque or semi-monocoque aircraft structures, in which all, or a substantial portion of the structure load is carried in the skin, have come into general favour among aeroplane designers. Beginning within the last 15 years, the stressed skin principle has also been applied to steel, aluminium, and magnesium parts, while the newer synthetic binding resins have been utilized to improve the wood-base structures of the pioneers of monocoque. Although certain plywood combinations would no doubt prove of the best structural value, owing to the limitations of the available supply of this material, it is necessary to use metals for stressed skin structures to a very large extent. These facts led the authors, early in 1940 to choose the field of metals in a research program directed toward obtaining more efficient stressed skin aircraft structures.

Steel.—Stressed skin structures of steel have been commonplace in other fields of endeavour but when designed to weight limits acceptable for modern aircraft, the result was almost always a comparatively thin sheet operating within its buckling range, and stiffened by a multitude of small formed beams, ribs, or stringers, spot welded or riveted to the main cover sheet. Here the cost of forming, handling, tooling and assembling becomes a serious if not prohibitive factor.

Aluminium.—Aluminium has become within the last 10 years an indispensable and major element in the designer's field of materials, and is used, reinforced by strips, extrusion beads, or ribs, for the major portion of the structure on most military and transport aircraft that fly to-day. Flat aluminium sheet, however, must be regarded as having a higher density than desirable, and is rarely used without some internal stiffening of strips or corrugations. Also while spot welding has been developed to an excellent degree of reliability, an exacting technique is required in its use, and many joints must be made where the physical limits of spot welding equipment do not permit its use. Consequently most modern aircraft contain from 100,000 to over 1,000,000 rivets which complicate manufacture and increase costs.

Magnesium Alloys.—Early in 1940, further investigation of available materials and methods of fabrication was carried out. Magnesium and its alloys soon proved to be most attractive. Less than two-thirds the weight of aluminium, and not much over one-fifth as heavy as steel, such materials have a relative stiffness, for a given weight of 2.5 times that of aluminium and 19.5 times that of steel. The facts that the production of magnesium is rapidly expanding, that the sources are inexhaustible, and that next to berylium it is theoretically the best possible material for simple metal monocoque structures, have assured its wide-spread use in aircraft.

Fabrication Methods for Magnesium.

Magnesium had previously been spot welded and gas welded successfully. However, rivets of magnesium alloy work-hardened so rapidly during driving as to prove impractical, so that other materials had to be used for rivets in the assembly of magnesium parts. Also, the ideal surface smoothness cannot be obtained by lap-joints, whether riveted or spot welded. Gas welding could only be accomplished under the protection of a heavy flux, due to the extreme affinity of magnesium for oxygen and nitrogen. And, unfortunately, the successful fluxes available were all of an extremely corrosive nature.

Finally, the search lead to arc welding which had previously been considered impossible on magnesium. The first experiments led to many small magnesium fires. An amazing number of preliminary experiments can be imagined, when all possible variations in alternating and direct currents, polarity, types and materials for electrodes, fluxes and blanketing gases were tried.

The first glimmerings of success occurred when the arc was struck between the work and a magnesium electrode which was supported in a hollow receptacle through which helium, under low pressure, was allowed to flow into the weld arc. With this arrangement, however, the control of the flow of material to the weld was erratic. Various refractory materials were then tried for the electrode, and when the research program reached the stage where a tungsten electrode was used in a helium atmosphere, success instantly crowned more than a year of experimentation and the Heliarc method of welding was born.

Helium Arc Welding of Magnesium.

Basically, this method of electric welding, useful with all standard direct current welding machines, consists in striking an arc between the work and a tungsten electrode, simultaneously feeding helium gas to the weld area through an annular nozzle surrounding the electrode, and feeding the additional weld material needed for the joint into the arc from an uncoated welding rod of substantially the same material as the work. Reversed polarity is used, i.e., the current flows from the work to the electrode. The flow of helium, fed to the work area at 0.25 to 0.5 lbs./sq. in., is controlled by a valve on the torch handle which is opened by the operator just before the arc is struck, and held open during the welding process. The arc is very quiet, there is no tendency to sputter or throw materials from the weld. This method of welding will probably find extensive use on alloy and stainless steels, where the results seem superior to those obtained by any other known method. The strength of the joint varies from 80 to 100% of the parent material, depending on the alloy and welding conditions, and there seems to be no limitation in the type of joint that can be made.

The helium blanket completely eliminates the use of any flux in the joint, and while minute quantities of tungsten are present in the joint, there are no adverse corrosive effects therefrom. Actually the weld appears somewhat more corrosion-resistant than the parent metal, there being a slight electrolytic balance which causes corrosion, if it appears at all, to be present in the sheet adjacent to the weld rather than in the weld itself. The seams are distinguished by their metallurgical purity, homogeneity, and absence of inclusions.

Design Considerations and Structural Details.

It was reasoned that the application of magnesium alloys to aircraft construction could be accomplished along two different principles. The first would be to design for maximum weight reduction. However, the undesirable physical properties of magnesium alloys (rapid strain hardening, corrosion, etc.) would probably limit the weight saving over a comparable aluminium alloy structure to about 10%, i.e., to some 3.5% of the empty weight of the airplane. It was, therefore, decided to favour the perfection of the aerodynamic shape and simplicity and low cost of structural construction.

The wing design was not made analytically by taking the weight of the present aluminium alloy riveted wing and reapportioning it to the various structural com-

ponents of the magnesium alloy wing. The Heliarc welded magnesium alloy wings were designed synthetically from the test experience and data already accumulated, and from the calculated loads acting on the wing.

Before the construction of the wings was begun, a very detailed weight analysis was made which indicated that the weight of the completed welded magnesium alloy wing structures should be approximately the same as that of the aluminium riveted wings; this had been confirmed by actual weighing of the finished structures.

The whole wing structure is composed of only two basic elements: The sheet, forming the monocoque shell; and extruded sections, forming the internal structure. The versatility of arc welded construction made it possible to limit the number of various extrusions to no more than four different sections. Furthermore, the preparation of the profile sections and sheets was greatly simplified, because flanges for riveting, and elaborate templates for the shaping of parts and the coordination of multitudes of riveting holes, were no longer necessary.

Manufacturing Problems.

Welded magnesium alloy constructions, too, are subject to shrinkage distortion. By making proper allowance in length for shrinkage this difficulty was solved as far as the dimensional control was concerned; for buckling distortion, a simple method was developed which consists of applying heat and pressure to the buckled structure through the use of ironing pads which relieve the internal strain in the sheet.

To make certain that no excessive locked-in strains are set up in Heliarc welded structures, experiments were carried out to obtain the absolute value of internal strains in magnesium alloys induced by welding. At the worst these stresses were found to be of the order of 1000 lbs./sq. in max., and are, therefore, of little consequence as far as the impairment of the integrity of Heliarc welded magnesium alloy structures is concerned. This is probably due to the relatively low modulus of elasticity and low yield strength of these alloys.

The amount of welding is not indiscriminate; proportioning of the welded seams to the loads carried through them and selecting the type of weld to best fit the conditions of elastic flexure of the structure should be two recognized principles of electric arc welding applications.

Serviceability of Magnesium.

In the past magnesium alloys have suffered from two generally known and popularly misunderstood faults. One is the general fear of their inflammability and the other is a deep-rooted conviction that these alloys corrode rapidly.

As to the first, the experience of the authors is that the fire hazard has been greatly exaggerated; in spite of the intensive welding development of these alloys in the shops during the last two years, the only fires involving magnesium were those started deliberately for test purposes, or in experiments before helium was used. It was discovered that the zinc chromate primer, generally used by the aircraft industry, acts as a potent fire inhibitor on magnesium, and that it is in fact impossible to ignite these alloys even artificially. Magnesium retains its elastic modulus to much higher temperature than is the case with aluminium alloys.

The weather durability of magnesium aircraft structures in service is still undetermined. However, a wealth of artificial corrosion testing, and also gratifying results of the use of magnesium alloys on several truck bodies through a number of years, furnish convincing proof that corrosion is not as dangerous as is generally believed, provided surface protection is given. This protection consists of treating the finished, welded and cleaned structures with sodium dichromate and painting them with standard zinc chromate primer and two coats of finishing lacquer. This protection has been found to be sufficiently elastic under load, as well as abrasion resistant.

One of the least desirable physical characteristics of the magnesium alloys is their inclination to strain corrosion. The elasticity of the surface finish helps here a great deal, but also care must be exercised in the design to avoid stress concentrations if possible.

Static tests of magnesium wings have demonstrated that these wings are elastically more flexible in bending than aluminium alloy wings. This is a desirable feature, as it tends to reduce excessive loads in gusty air, particularly when it is realized that the internal damping of magnesium alloys is several times greater than of aluminium alloys. On the other hand, the magnesium wings are more rigid in torsion than corresponding aluminium alloy riveted wings. This is also a very desirable property to eliminate danger from flutter, and can be traced to greater thickness, and to the absence of slippage in welded seams.

Cost Comparison.

The evaluation of the economic advantages of electric arc welding as applied to magnesium aircraft structures is a difficult task because of the many variables involved. A direct comparison of the cost of a Heliarc welded seam and a riveted seam in the same materials shows that the total cost per foot is 1.14 dollars for the welded structure and 1.34 dollars for the riveted structure. Such a comparison, however, is unduly conservative in that the joining cost is only a minor element in the overall economic gain to be made. If the welded magnesium structural design is directed primarily to simplicity and aerodynamic excellence, it will be found that weights generally equivalent to those of contemporary structures of riveted aluminium alloy will be obtained. On the other hand, the reduction in number of parts for a given structure and the possible reduction of drag of the finished airplane are factors of great importance. In addition, the actual number of pieces required in the design is in the order of one half the number required in the comparable aluminium alloy structure, so that the cost of fabrication may be expected to be reduced in a similar measure. Unfortunately, at this writing the welded wings are only being produced in experimental quantities and no cost data on conventional wing structures in comparable quantities are available to the authors.

General Economic Evaluation.

The actual structural cost in itself is still of minor importance in the overall economic advantage to be gained, however, because the most valuable contribution of this program lies in the possible reduction in drag of the finished airplane. Monocoque welded magnesium structures are designed with comparatively thick skins which do not buckle locally within the normal range of flight loads. Their surface finish can be as smooth as that of a fine automobile, and held within accurate limits. The outer surface of all welded joints may be ground flush with the face of the surrounding sheet. Depending somewhat on cover thickness and internal structure, some slight surface irregularities may exist, but these at the worst can be limited to long waves of very low magnitude which do not adversely affect the drag of the structure. In summarizing this point it may be said that a parasite drag reduction of at least 30% may be obtained through the use of the new low drag airfoils which, to the best of the authors' knowledge, can only be built to proper accuracy and finish in Heliarc welded magnesium, if metal is to be used.

POWDER METALLURGY

PHYSICAL PROPERTIES OF PARTS MADE FROM IRON POWDER.

By F. V. LENEL, Moraine Products Division, General Motors Corporation, Dayton, Ohio. (From *Mechanical Engineering*, Vol. 65, No. 7, July, 1943, pp. 489-492).

THIS paper will not deal with such products as refractory metal wire and sheet, cemented carbide tools, self-lubricating bearings, electrical-contact materials, etc., whose manufacture is reserved almost exclusively to powder metallurgy. It is concerned with those products which can be made by conventional methods, such as die casting or machining of wrought or cast metal as well as by powder metallurgy methods.

By briquetting and sintering a product to the finished size, the cost of machining and the cost of scrap produced in machining are saved. In order to make the powder metallurgy process competitive, these savings must compensate for the increased cost of raw material, that is, powder versus bar stock or casting, and the cost of briquetting, sintering and sizing. In some cases, where only semi-finished product can be briquetted, the cost of the final machining will have to be added to the powder metallurgy operations in order to arrive at comparative costs. A complete comparison of cost, based upon the number of pieces to be produced, should also include the cost of tooling for briquetting and sizing, as against the tooling set-up for machining by conventional methods.

Reference will be made particularly to those products which are manufactured from iron powder, and which, therefore, compete with products machined from steel bar stock, steel forgings, or steel or cast-iron castings. In the past many claims have been advanced with regard to the mechanical properties of such iron parts ; and it has been felt throughout the industry that a clarification is necessary as to what mechanical properties can be obtained consistently in large-scale production.

The manufacturers of metal powder products have organized a committee to prepare general specifications for iron parts made from metal powders. The formulation of these specifications has met with problems which are peculiar to the powder metallurgy process and which should be thoroughly understood not only by the manufacturer but also by the user of powder metallurgy products, the size and shape of which has a direct bearing on their mechanical properties.

The tensile strength, ductility, hardness, impact, strength, etc., of a finished product machined from bar stock are assumed to be the same as those of the stock from which the piece was made. In castings, coupons can be cast together with the casting, the mechanical properties of these coupons can be determined, and in this way representative values for the mechanical properties of the casting can be obtained, although the influence of the section thickness must be taken into consideration. In powder metallurgy products, the effect of size and shape of the finished product upon its mechanical properties is even more accentuated.

MECHANICAL PROPERTIES.

Metal powder products are made by briquetting metal powders in steel dies and sintering the briquetted shapes in furnaces with controlled atmospheres. After sintering, the products may be further processed by a sizing operation or by sizing and subsequent resintering. For purposes of this discussion, the mechanical properties of the finished pieces may be described as depending upon four factors, as follows :

(1) Composition of the powder mixture
(2) Grade of powders used
(3) Temperature, length of time, and atmosphere of the sintering and resintering operations
(4) Density, resulting from briquetting, sintering, and sizing operations.

The first three of these factors have parallels in cast and wrought materials (chemical composition, purity, heat treatment) but factor (4) is peculiar to metal powder products. The density depends not only upon the processing of the product but also upon its size and shape. This correlation between density, processing conditions, size and shape of the product, and mechanical properties may be treated under the following headings :

1 How the mechanical properties depend upon the density of the product
2 How the density depends upon the processing steps
3 How the density depends upon the size and shape of the product, and
4 How the size and shape of the product influence its processing.

1. The tensile strength, impact strength, and hardness, as well as the ductility, of a metal powder product will increase with increasing density of the products, if the same powder mixture, quality of powder, and sintering treatment are used (Fig. 1).

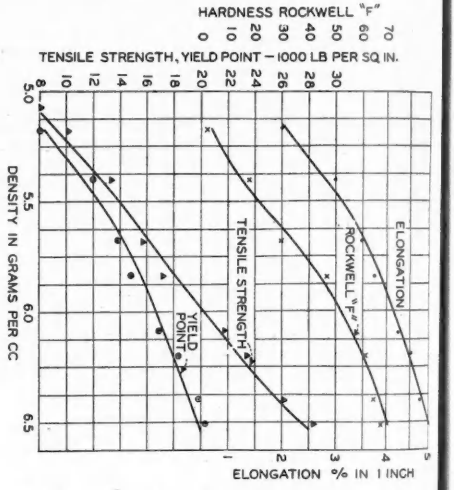

HARDNESS ROCKWELL "F"
0 10 20 30 40 50 60 70
TENSILE STRENGTH, YIELD POINT — 1000 LB PER SQ IN.
8 10 12 14 16 18 20 22 24 26 28 30
DENSITY IN GRAMS PER CC
5.0
5.5
6.0
6.5
YIELD POINT
TENSILE STRENGTH
ROCKWELL "F"
ELONGATION
0 1 2 3 4 5
ELONGATION % IN 1 INCH

Fig. 1. Relation of mechanical properties to density for test pieces briquetted from a hydrogen-reduced iron powder and sintered 3 hours at 2000 °F.

2. The density of a metal powder product will depend to a certain extent upon the powder mixture, the quality of the powder, and the heat treatment of the product, but most important will be the briquetting pressure used to compact the powder into a coherent piece, and the sizing pressure used in coining or cold-forging the product after it has been sintered. A piece having a given size and shape will be the denser the higher the briquetting pressure and the higher the sizing pressure. Fig. 2 shows the correlation between briquetting pressure and density.

3. Due to the fact that most metal powder products are briquetted only along one axis, the density of pieces of different shapes and dimensions will be different even if the same briquetting and sizing pressure are used. Therefore, the friction between the die and the powder

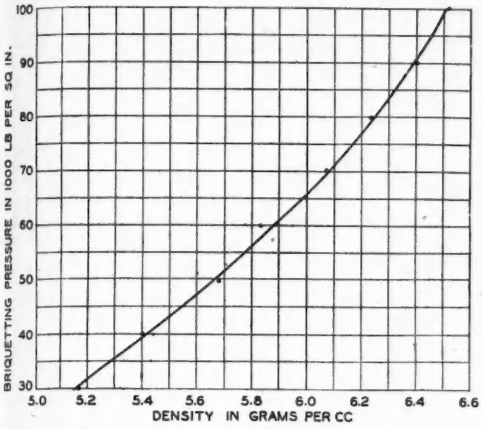


Fig. 2. Relation of density to briquetting pressure for test pieces briquetted from a hydrogen-reduced iron powder and sintered 3 hours at 2000 °F.

will influence the density in two ways :
(a) With a given briquetting and sizing pressure, pieces which are long and slender in the direction of pressing will be less dense than pieces which are short and thick (Fig. 3) ; (b) Pieces whose ratio of length (dimension in the direction of pressing) to width (dimension perpendicular to direction of pressing) is large (in general greater than 2) will have an uneven density over their length. If pressed from both ends they will be denser at the ends than in the middle, and if pressed from only one end, they will be denser at one end than at the other (Fig. 4).

Pieces which have a nonuniform section in the direction of pressing (e.g., flanged pieces) may either be made with single punches or with multiple punches sliding relative to each other. If the piece is made with single punches the density will be higher in the short section than in the long section, because of the difference in the rate of compression of the powder. For some applications this difference in density and the corresponding difference in strength, ductility, and hardness is permissible ; for other pieces multiple punches are necessary to make the density throughout the section as even as possible (Fig. 5).

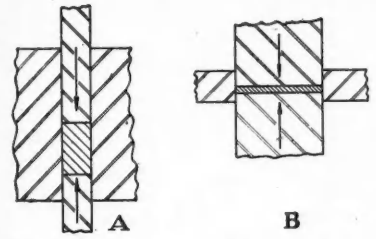


Fig. 3. Schematic sketch illustrating lower density of long slender piece A, as compared with short, thick piece B.

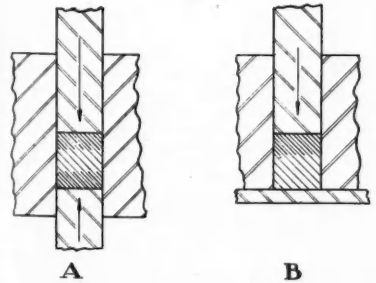


Fig. 4. Schematic sketch illustrating uneven density distribution in pieces briquetted from both ends A, and from one end B.

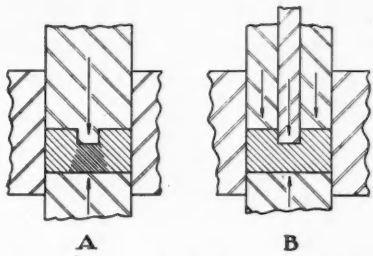


Fig. 5. Schematic sketch illustrating uneven density distribution in a piece with non-uniform section when briquetted with a single punch A, as compared with the even density distribution in the same piece when briquetted with multiple punches B.

4. Briquetting and sizing pressures are limited by the pressure which the dies will stand without rapid failure by fatigue or wear. They will also depend upon the shape of the piece and the design of the die. For large pieces, the capacities of the available briquetting and sizing equipment may also be limiting factors. Complicated pieces have, therefore, lower density than standard test pieces. In some cases, the lower physical properties, due to this lower density, may be compensated for by a change in the grade of powder or the sintering conditions, but such changes will also alter the cost picture.

Three Classes of Metal-Powder Products.

In order to grade parts made from iron powder, according to their properties, they may be divided into three types as follows :
Type A : Materials having mechanical properties similar to common cast iron suitable for applications where the stresses are very low.
Type B : Materials similar to type A, having improved tensile strength, a definite yield point, and a noticeable elongation.
Type C : Materials having mechanical properties approaching ordinary malleable iron, suitable for applications where stresses including impact are moderate.

The grade of powder used, the briquetting and sizing pressures, the time and temperature of heat-treatment would determine in what class a particular product would fall, changes in one or more of these factors possibly shifting a product from one class to another. The three types would be identified by their specifications for chemical composition and for density, as suggested in Table I.

TABLE I. CHEMICAL AND DENSITY REQUIREMENTS.

TYPE	Total carbon, per cent	Total iron, per cent	Density g per cc
A	2.5 (Max)	95 (Min)	5.4 (Min)
B	0.4 (Max)	97.5 (Min)	5.8 (Min)
C	0.2 (Max)	98.5 (Min)	6.5 (Min)

The table shows that for parts with higher strength and ductility, an impure product with low density would not be suitable.

Type A would include parts which contain up to 2.5% carbon which may be added to the powder mixture in the form of graphite and would combine with the iron during the sintering operation. Such carbon-containing iron-powder products are harder and stronger than straight iron products, but they are also more brittle.

STRENGTH TESTS FOR INDIVIDUAL PARTS.

Because a general specification has to be broad, it will be necessary to develop detail specifications for individual parts. They should be set up after performance tests on the parts have shown that they are satisfactory for the application. The detail specifications should furnish a check on the uniformity of the production and, whenever possible, on the performance characteristics of the tested product. Such tests are: (1) Crushing, radial or shear strength, (2) Bending or impact drop test, and (3) Hardness at one or more specified points of the piece or hardness variation over the entire piece.

Static strength tests would be suitable for pieces where no particular impact resistance is needed, e.g., for oil-pump gears. Holding the gear in a fixture the load required to break off one of the teeth is determined. Hollow cylindrical pieces may be tested by a radial break test; the piece is crushed between two flat plates and the load is measured when failure occurs. A bending or impact test should be devised to determine the shock resistance. A material which does not show any elongation in a tensile test, or no bending in a bend test, will usually be quite shock-sensitive. On the other hand, a material which has a fair amount of ductility will not break so easily under a sudden blow. Fig. 6 shows a piece requiring shock resistance, and the fixture designed to test the piece. The piece is clamped in the vice in such a way that one side of the slot is flush with the edge of the vice, with slightly more than one half of the piece protruding from the vice. The specification would read that a specified weight dropped from a specified height should not fracture the piece and that the same weight dropped from some lower height should not cause a crack in the piece.

Fig. 7 shows another specimen requiring a certain amount of shock resistance. It is briquetted with the long section in the direction of pressing, pressure being applied both from the top and the bottom. It will show, therefore, some difference in density along the long section. It will have the lowest density and, therefore, the lowest strength and ductility in the middle of the long section. The fixture is designed to test the piece at its weakest point, the idea being that if the piece does not fracture at this point it will be satisfactory throughout. Simultaneously, the point where the flangelike section and the long section join is tested by bending the flangelike section. The actual piece is not expected to be bent in service but it will have to withstand a certain amount of shock, and the best test will also give a good indication of the resistance to shock of the piece.

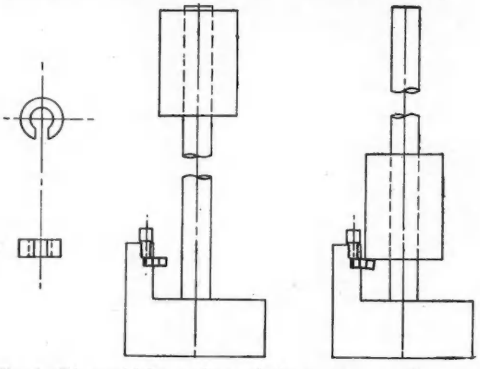


Fig. 6. Piece which may be made by powder metallurgy, and a fixture designed to test its shock resistance.

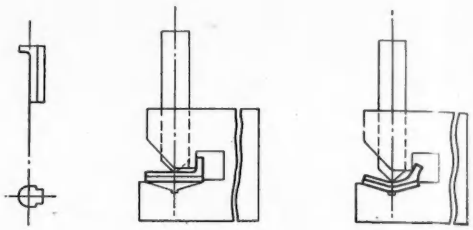


Fig. 7. Piece which may be made by powder metallurgy, and a fixture designed to test its ductility.

Both tests described are destructive tests, in many applications, however, no nondestructive test would be satisfactory.

HARDNESS TESTING.

Indentation hardness tests are very convenient and may be useful for powder metallurgy products, but understanding of the mechanism of an indentation test on porous material is absolutely necessary. The resistance to indentation, which is measured, is a function not only of the chemical composition and the microstructure of the tested piece, but also of its density. The lower the density, or in other words, the less material there is to resist the deformation, the further will the indenter sink into the material and the lower will be the reading. Two pieces which show identical hardness readings may, therefore, have entirely different properties. One may be low in combined carbon, but high in density and, therefore, quite ductile and shock-resistant. The other may have a high combined carbon content but a low density and would, therefore, be quite brittle and shock-sensitive.

The influence of density on the indentation-hardness reading must also be considered when the abrasion resistance of a material is to be evaluated. The abrasion resistance depends much more upon the microstructure of the material than upon its density. For instance, a piece made of iron powder and containing sufficient combined carbon may be heat-treated and may then be fully martensitic. Such a piece will be file-hard, but the hardness reading on the Rockwell C scale may be only 30.

The tensile properties of products made from powder should not be included as part of a general specification; however, such properties can be and have been measured. The values shown in Table II. were determined on special subsize specimens. Typical values are given for the three previously suggested types of material. The values given in Table II., which are purposely on the conservative side, are not very impressive. Under laboratory conditions, it has been possible to obtain much superior properties, but no large-scale production technique has been developed which would permit the manufacture of parts from iron powder with high strength and high ductility at a competitive cost. The main difficulty lies in achieving satisfactory ductility and impact resistance, while it is much less difficult to obtain higher strength and high hardness by proper heat-treatment. These heat-treated materials are not specifically included in Table II.; they are rather brittle and cannot be used in any application where they have to withstand shock.

TABLE II. TENSILE PROPERTIES.

Type	Tensile strength, p.s.i.	Elongation, per cent in 1 in.
A	15,000	.5
B	25,000	3.0
C	35,000	7.0

Wide application to Small-Arms Parts.

In many components of small arms or of expendable munitions, to mention only two fields which are at present in the foreground of interest, an accurate configuration is much more important than high mechanical properties. Many of these components are at present made from low-carbon steel because that is the cheapest and most convenient material available. However, that does not mean that the physical properties actually needed are those of low-carbon steel. These are the applications where powder metallurgy is believed to have a field, if by this method a part can be made at a competitive cost and with a saving in machine time and in bar stock. The metal-powder-parts manufacturer and the user of the parts should co-operate in selecting suitable applications and in setting up detail specifications which will insure the quality and uniformity of production.

MODULATOR CONTROL FOR MULTI-FUEL FIRED FURNACES

By C. J. GEISER, Republic Flow Meters Company, Chicago, Illinois. (From "*Iron and Steel Engineer*," Vol. 20, No. 8, August, 1943, pp. 35-39).

IN steel plants many furnaces are equipped to use two or more fuels simultaneously and in combination, depending upon availability, respective costs, or furnace operation. The automatic control of combustion air on any furnace fired simultaneously with several fuels, has always involved the problem of summarising the fuel flows in order to obtain an index for airflow control. The difficulty to be faced in designing the summarising device lies in the fact that it must summarise linear functions (fuel flows), while the index of each function is a square root function (flow differential). This square root function is involved in all measurements of fluid flow by means of a thin plate orifice, flow nozzle, Venturi tube, etc.

A novel and simple means of correctly summarising the flows of two or more fuels is the modulator consisting of an inclined U-tube operating on a force balance principle. Fig. 1 illustrates the square root modulator equipped with a diaphragm type measuring element which is sensitive to the input loading air pressure, or flow differential. The force transmitted from the diaphragm element to the U-tube assembly serves to unbalance the tube, hereby controlling the air bleed at the multiplying valve. The resulting air pressure at the multiplying valve is piped to the outer leg of the U-tube, and the resultant travel of mercury within the tube produces a force which balances that exerted by the measuring diaphragm. Thus the resulting air pressure from the multiplying valve is a function of the input loading pressure, or differential, and this resulting air loading pressure is the output static loading pressure from the unit. The relationship of these pressures is that the output pressure is proportional to the square root of the input pressure or differential.

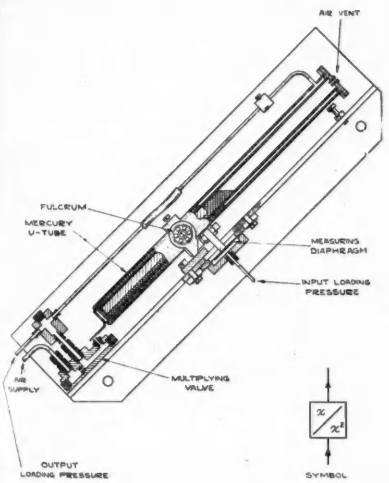


Fig. 1

With the use of square root modulators on multi-fuel applications, static loading pressure directly proportional to each fuel flow is established. Being linear, these static air loading pressures may be directly totalised on a diaphragm type pressure totaliser (Fig. 2), the output static loading pressure of the latter being used as an index of the total air requirements of combustion air.

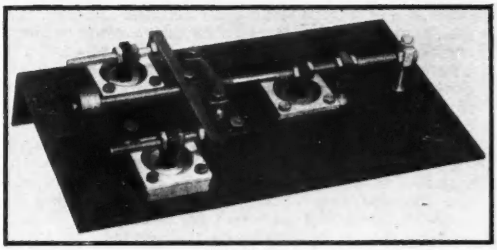


Fig. 2

However, before the pressure is used to control the combustion air flow regulator, the square function must be established in order to achieve a correct balance against the air flow differential. This is accomplished with the use of the squaring modulator, shown in Fig. 3, which is an inverse arrangement of the square root modulator. The output static air pressure is therefore proportional to the square of the input pressure.

An arrangement for the automatic control of the

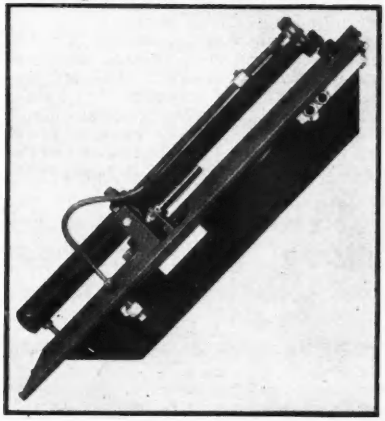


Fig 3

combustion air in conjunction with the manual control of two fuels is shown in Fig. 4. This type of control is common on open hearth and other types of furnaces where automatic temperature control is not practical or desirable. The flow differential of each fuel flow is measured by the differential diaphragm of a square modulator, its putput pressures being directly proportional to the respective firing rates. To enable correct adjustment of a multi-fuel control for changes in the calorific value in the specific gravity, and also in the air requirements of the fuels, means for adjustment must be provided which are independent for each fuel. A convenient method to effect this correction is embodied in the modulator type of multi-fuel control.

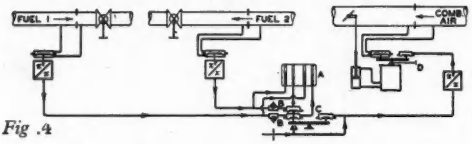


Fig .4

In Fig. 4, the first and third tubes of the mercury monometer gauge "A" indicate the prevailing firing rates of the fuels; while the ratio bleeder valves "B" provide for individual adjustment of the loading pressure for the air requirements of each fuel. The adjusted pressures are shown at the second and fourth monometer tubes. The output static pressures from valves "B," which are linear functions proportional to the required combustion air flow, are totalized at totalizer "C." The resultant loading pressure, being a linear function of the total air requirements, is in turn impressed on a squaring modulator which produces the correct characteristic for combustion air control.

While a system of this kind represents the correct method for the adjustment of the fuel-air ratio, the system always encounters obstacles because of the fact that the adjustments required must be effected by the operating personnel. Correct adjustment of the fuel-air ratio would always require the provision of an independent ratio-setting device for each fuel fired, and the additional adjusting devices involved are usually beyond the practical use of the first helpers and heaters on steel mill furnaces. For this reason, a single fuel-air ratio adjustment "D" has been provided with a calibrated setting indicator. This single ratio adjustment will not be correct whenever the measured orifice differentials do not represent the quantity and quality of fuels for which the control was set. These limitations must therefore be weighed against the ability and willingness of the operator to carry out successfully the required adjustment of the individual fuel-air ratios.

In Fig. 5, automatic control is extended to fuel No. 1, in addition to combustion air control, while manual control of fuel No. 2 is retained. This arrangement also contains a flow control subpanel "E" for manual control of fuel No. 1, whenever required. Similarly, this fuel control principle can be extended to two or more fuels. Referring to Fig. 5, it is seen that the use of individual fuel-air ratio valves "B" does not interfere with the correct operation of the fuel flow control. The loading pressure sent out from subpanel "E" and passing through the modulator to the fuel flow regulator "F" is not subject to adjustment for obtaining the correct fuel-air ratio.

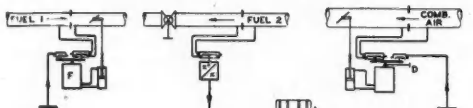


Fig. 5

Compensation of the modulator system for air infiltration is provided for, hereby permitting a highly flexible adjustment to almost any characteristic that may be called for. This compensation is effected by the appropriate weight loading of the weigh-beam of the totalizer and/or of that of the combustion air controller. As the static air loading pressures at the totalizer are in linear relationship with the air flow, the weight loading of the totalizer weigh-beam affords a correction for a constant amount of air infiltration into the furnace. On the other hand, where an increasing or decreasing amount of air infiltration with the firing rate is to be compensated for, this can be effected by appropriate loading of the weigh-beam of the combustion air regulator. In this way, the correct combination of unbalanced weight at the weigh-beams of these two regulators makes it possible to obtain practically any characteristic that may be desired. Furthermore, it is possible to establish a minimum rate of air flow such as may be required by the operating characteristics of the fuel burners used. This measure, which may also be called for in the operation of regenerative furnaces, is realised by the employment of a minimum pressure setting relay which is connected to the output pressure line of the totalizer.

THRESHOLD TREATMENT OF COOLING WATER BY SODIUM-HEXAMETAPHOSPHATE.

By DR. ING. E. SCHUMANN and P. LANDERS. (From "*Die Wärme*," Vol. 66, No. 14, April/May, 1943, pp. 122-128).

THE threshold treatment of condenser-cooling water by means of hexametaphosphate had been carried out for the first time in Germany in 1939 at a central station in the Ruhr district. By October, 1942, its application had been extended to 31 power plants in the Ruhr district alone. This threshold treatment rests on the observation that the addition of a very small amount of hexametaphosphate to the cooling water containing calcium bicarbonate will prevent the formation and precipitation of calcium carbonate under the influence of temperature increases, pressure decreases, diminution of the partial pressure of the carbon dioxide and its decreased solubility in water, etc.

Stable conditions were found to obtain with a Ruhr water of 14-15 German degrees of carbonate hardness when containing 2 p.p.m. hexametaphosphate. With this method of water treatment, no difficulties with scale deposition occurred. It is, however, necessary to continually drain a certain quantity of water from the cooling circuit and to replace it. Since the cold water thus added increases the condenser vacuum, the available heat drop is also increased. In certain plants in the Ruhr district, the drain from the cooling circuit is limited to a point where the concentration of the circuit water amounts to twelve times that of the original water supply, hereby considerably reducing the drain from the system. No scale formation could be observed with this mode of operation. There was, however, a certain sludge formation which, from the viewpoint of heat transfer, was considered objectionable. For this reason, the condenser tubing of certain plants is periodically cleaned by blowing compressed air through the condenser while the latter is kept in operation. The increase in water velocity caused by the admission of the compressed air was found to bring about removal of the sludge.

It was found that in order not to stimulate the formation of sludge, the maximum amount of hexametaphosphate *actually* contained in the cooling water must not exceed 2 p.p.m. Further tests showed the concentration of the phosphate solution in the storage containers to be of decisive influence upon the sludge formation, and should not exceed 0.1 per cent. The tanks used for the preparation of the solution must therefore be of ample proportions. If the tank available is not large enough to hold a 24 hour supply, new solutions must be prepared at the required intervals. The previous practice of introducing the required amount of solution on the suction side of the circulating system has now been superseded by injecting the solution into the delivery line of the circulating pump.

TECHNICAL NEWS

Announcements in this section include News relating to British Firms, and on Equipment produced by British Manufacturers.

Available literature may be secured by addressing a request to the advertising department of "The Engineers' Digest," or by writing direct to the manufacturer and mentioning "The Engineers' Digest" as a source.

- ✦ NEW EQUIPMENT
- ✦ BOOKS AND CATALOGUES
- ✦ BUSINESS CHANGES
- ✦ PERSONAL
- ✦ NOTES

PERSONAL.

Mr. J. Benstead and **Mr. E. Thornton** have been appointed Members of the Advisory Council to the Committee of the Privy Council for Scientific and Industrial Research.

Mr. H. Boot has been appointed assistant General Manager of the Consett Iron Co. Ltd.

Sir George Campbell has been appointed Regional Port Director for the South-East of England, including the Port of London.

Sir William Y. Darling has joined the Board of Bruce Peebles & Co. Ltd.

Sir I. Fortescue Flannery, M.Inst.C.E., Hon. M.I.Mech.E., M.Inst.N.A., former President of the Institute of Marine Engineers, the Liverpool Guild of Marine Engineers and Naval Architects, and the Junior Institution of Engineers, has died recently at the age of 92. He was the founder and Chairman of Flannery, Baggaley and Johnson Ltd., Chairman of Callender's Cable and Construction Co. Ltd., and of W. and F. Avery Ltd., from which position he only resigned at the beginning of the war.

Dr. W. Hatfield, Director of Research at Thos. Firth and John Brown Ltd., and Director of Firth-Vickers Stainless Steels Ltd., died on the 17th October, a day after his presidential address to the Sheffield Society of Engineers and Metallurgists had to be cancelled because of his illness.

Dr. Hatfield was Vice-President of the Iron and Steel Institute, Chairman of the Technical Advisory Committee of the Special and Alloy Steels Committee formed by the Ministry of Supply, and Member of the Iron and Steel Research Council. In 1935 he was elected a Fellow of the Royal Society.

Dr. Hatfield has given many lectures and read many papers before scientific societies and technical institutions, and he was one of the best known British metallurgists in this country as well as in the United States where he was in 1928 as Campbell Memorial Lecturer at Philadelphia.

Dr. A. H. Langley, B.Eng., A.M.I.E.E., has been appointed Head of the Electrical Engineering Department of the Sunderland Technical College.

Dr. W. A. Macfarlane has been appointed Director of Fuel Efficiency in the Ministry of Fuel and Power.

Mr. S. B Turner has been elected Chairman of the London Section of the Institute of the Plastics Industry for 1943-44.

AUTOMOBILE RESEARCH COMMITTEE.

At the quarterly meeting of The Automobile Research Committee of The Institution of Automobile Engineers, held at Caxton Hall on October 13th, Mr. John Shearman, M.I.Mech.E., M.I.A.E. (Road Motor Engineer of the L.M.S. Rly.) was re-elected Chairman of the Committee for the year 1943-44, and Mr. A. G. Benstead, M.I.A.E (Rotax Ltd.) was re-elected Vice-Chairman.

The election of Sub-Committees dealing respectively with research and with general administration was carried out, and reports were presented showing the progress of researches at present in hand and the growth in the membership, the number of affiliated firms now being 364.

Lt.-Col. J. A. Cole, who recently retired from the position of Managing Director of Humber Ltd., and who had represented that firm on the Committee since its inception was co-opted as a member in order that the Committee might continue to have the benefit of his services.

BRITISH ENGINEERS' ASSOCIATION.

At the 31st annual general meeting of the British Engineers' Association Mr. Cecil Bentham was re-elected President of the Association. Lieut.-Colonel H. B. Rigall, of Ruston and Hornsby Ltd., and Mr. W. D. Lancaster, of Hughes and Lancaster Ltd., have been elected Vice-Presidents during the year.

BRITISH INSTITUTION OF RADIO ENGINEERS.

Sir Louis Sterling presided at the Annual General Meeting of the British Institution of Radio Engineers, held on Friday, September 2nd, in the Lecture Hall of the Institution of Structural Engineers, 11 Upper Belgrave Street, London, S.W.1.

At the conclusion of business the president welcomed Sir Ernest Fisk, who, he said, pioneered radio development between this country and Australia.

Sir Ernest, addressing the meeting, made a most lucid and interesting survey of the history of wireless telegraphy and broadcasting from the beginning of the century, describing some of the technical and prejudicial difficulties that had to be overcome before direct Empire communication, so essential to Australia, could become an accomplished fact.

CATALOGUES RECEIVED.

Bostik Products Instruction Manual.—The B.B. Chemical Co. Ltd., Ulverscroft Works, Leicester, has issued an Instruction Manual for operations in the aircraft and other industries using Bostik products as glazing compounds, sealing compounds, adhesives, and cleaners. The manual is intended for Production, Design and Planning departments, while a Process Chart, a quarter-sized reproduction of which is included in the manual, is intended for exhibition in workshops where the actual operations are being carried out.

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BRITISH STANDARDS.

(Copies of British Standards may be obtained from the British Standards Institution, 28 Victoria Street, London, S.W.1).

BRONZE OIL RETAINING BEARINGS B.S. No. 1131

THIS standard has been prepared in collaboration with the Ministry of Supply and the three manufacturers of this type of bearing. The primary object underlying the preparation of the standard is to bring about a modification in the variety of types and sizes of bearings, and so to facilitate more rapid and economical production by concentrating the supply and demand on a simplified range of standard sizes.

The present standard relates to cylindrical type bearings of nominal bores from 3/16 inch to 2½ inches. Various alternative standard lengths are established in the schedule and the appropriate limits are given for the bore of the housing into which the bearing is to be assembled. Each size and length of standard bearing has been given a specific reference number for convenience when ordering. Further specifications are to be issued later for flanged bearings and thrust washers. Price 2/-.

P.D. 144 and P.D. 145.—AMENDMENT SLIPS TO B.S. 970 and 970b RESPECTIVELY.

AN amendment slip to B.S. 970 has just been issued by the B.S.I. under the reference P.D. 145, which provides for a revised Specification for Steel En. 100 and a new Steel En. 110. The amendment indicates that this steel which is primarily for bars for machining may also, by agreement between purchaser and manufacturer, be used for bars for forging and drop forging. It also provides for a new Specification for a steel with low contents of nickel, manganese and molybdenum. This steel has been standardised in order to utilise to the greatest advantage the low content of steel scrap. The material is intended as a substitute for steels to Specifications En. 16, 17 and 19, and covers ranges from 45-70 tons per square inch.

In view of these additions that have been made to the schedule, certain alterations were necessary to the direction issued by the Iron and Steel Control with regard to steels in B.S. 970, and these modifications are covered by amendment slip P.D. 145. Free.

B.S./STA. 19.—GILDING METAL STRIP.

THE above is a further Specification in the series relating to Non-Ferrous Metals which is being issued by the B.S.I. for the Ministry of Supply in order to co-ordinate the requirements of various departments of the Ministry, and also those of the Admiralty.

The Specification covers gilding metal strip used for bullet envelopes and includes requirements of the chemical composition which is set out in detail, and tolerances on the dimensions of the strip. There are two hardness ranges of material covered by this Specification, and separate bend test requirements are specified for each range. Requirements for grain size are also specified. Price 6d.

P.V.C. CABLES FOR USE IN SHIPS.

THE B.S.I. announces the publication of a Supplement to B.S. 883, Cables for use on Ships, dealing with the use of P.V.C. cables as an alternative to rubber-insulated cables. This Supplement follows closely the lines of the Supplement recently issued in connection with B.S. 7. Rubber-insulated Cables for general use.

The Supplement is contained in P.D. 149, which also sets out a number of amendments to the Specification for rubber and paper-insulated cables for use in ships, these changes being necessary to meet the present position regarding the availability of various materials. Price 2/-.

P.D. 157. AMEND. No. 1, SEPT., 1943, TO B.S. No. 138-1935.

THE B.S.I. has just published a War Emergency amendment to B.S. 138-1935 Portable Chemical Fire Extinguishers. This is a wartime amendment and provides for the use of steel in place of brass in the manufacture of the acid bottle cage. Free.

SPECIFICATIONS Nos. 12 and 12a, COPPER RODS FOR RAILWAY ROLLING STOCK.

THE B.S.I. has just issued an addendum slip to the above British Standard (B.S. 24, part 5) in order to lay down the dimensional tolerances to which the diameter of material supplied to this specification should comply. The addendum slip bears the reference number P.D. 158. Free.

BRITISH STANDARD SCHEDULE OF CAST IRON DRAIN FITTINGS, SPIGOT AND SOCKET TYPE (B.S. No. 1130).

THE B.S.I. has just issued a British Standards No. 1130 Schedule for fittings for Cast Iron, Spigot and Socket drain pipes. It was undertaken at the request of the Ministry of Works, and indicates the range of standard cast iron drain fittings, to be used with the pipes referred to in B.S. 437.

This Schedule covers such fittings as access and inspection fittings, traps, raising pieces, diminishers, gulleys, gratings, inlets and outlets, etc. It is hoped at a later date to give the essential dimensions of each fitting. Price 2/-

RUBBER AND INSERTION JOINTING FOR FLANGE AND SIMILAR JOINTS.

A WAR EMERGENCY revision of B.S. 945 Rubber and Insertion Jointing for Flange and Similar Joints has recently been issued. This revision has been made necessary by the urgent need to conserve rubber and supersedes B.S. 945-1941.

Certain relaxations in the quality of rubber have been made, and the Specification is restricted to three types—Soft, Medium and Hard, in place of the five types covered by the earlier edition.

The Specification is more comprehensive in one respect, in that it has been extended to cover insertion and sheeting used for high and low pressure steam, as well as for water pressure at normal atmospheric temperatures to which the original Specification was confined. Price 2/-

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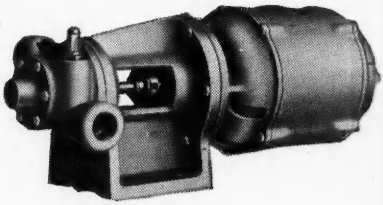

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